COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

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Semi-outdoor station of the Utah Power & Light Company under construction

Semi-Outdoor Station Placed in Service

Some Recent Developments in Operation and Maintenance of Surface Condensers

COMBUSTION

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

VOLUME EIGHT NUMBER TWO

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FOR AUGUST 1936

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EDITORIALS

DEPARTMENTS

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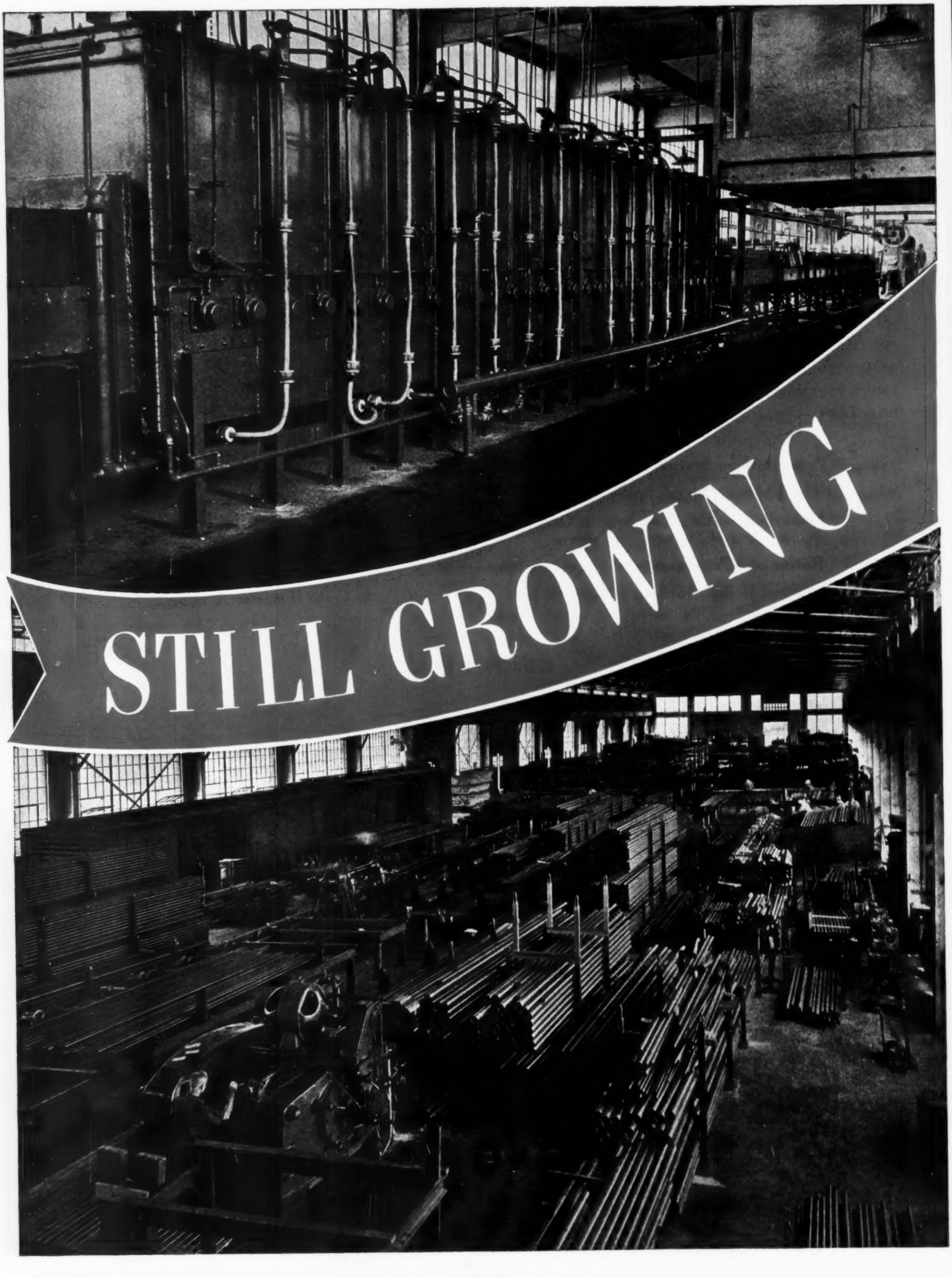

STILL GROWING

EDITORIAL

Utility Accidents

The National Safety Council has just issued a report on accidents in the public utility industries. While the severity rate, probably because of line fatalities, exceeds the average for thirty major industries reporting, the frequency rate is considerably lower—10.2 compared with 14.02, this rate representing the number of disabling injuries per million man-hours worked.

Advancement in safety, through educational campaigns, adherence to safety regulations and improvement in protective equipment, showed most gratifying results from 1926 to 1935, particularly among the utilities which led all other industries by a wide margin. They showed a 66 per cent reduction in the severity rate and 79 per cent in the frequency rate during this period, and as evidence of a continuing downward trend the former was reduced five per cent and the latter one per cent in 1935 over the figures for 1934.

Electrical equipment was the principal agency of injury and linemen were involved in more than half of all the injuries. Chance taking, disregard of instructions and other wrong attitudes on the part of the employees figured in a large number of the cases.

Despite the most rigid precautions that a company may adopt the human element is ever present and is often most difficult to cope with, particularly among experienced employees. The most any company can do is to install safety devices wherever feasible, prescribe definite operating procedure and instill the safety idea into its employees to the best of its ability.

The best known all-time "no-injury" record among utilities was held by the Nebraska Power Company from July 28, 1931 to Sept. 8, 1934, while the Iowa-Nebraska Light & Power Company had the best record for 1935 among large organizations, it not having had a single disabling injury out of 1,687,000 man-hours. Several other companies had very creditable records.

Vocational Guidance

That there are many misfits in engineering, as well as other professions, has long been apparent; but the number is small compared with the host of individuals who have taken up engineering studies and dropped by the wayside, either voluntarily or through inability to keep pace with the instruction. Many boys are permitted to take up engineering courses without any conception on their part, or on the part of their parents and high-school teachers, of what is involved in its many branches, the qualifications for success and the opportunities offered. Often a boy will get the notion that he wants to take up engineering because some chum has so matriculated.

One of the several objectives of the Engineers' Council for Professional Development is vocational guidance, and it is encouraging to note that a tangible start has been made in Detroit. There, at a recent conference which was attended by 220 people, including 123 high-school boys, prominent engineers, actively identified with industry, explained the different fields of engineering and the vocational activities within them, such as design, research, sales, etc. No attempt was made to "sell" engineering to those present, but rather to afford them a better understanding of what it encompasses.

Following this, the Conference divided into several groups for those interested in particular branches of engineering where councillors representing the several engineering societies answered questions and explained the subjects in greater detail. The Conference had the support and active cooperation of the Detroit Board of Education, the local technical societies and industrial groups. Plans were laid for further consultation with students and parents.

It is to be hoped that engineering groups in other localities will emulate the good work started in Detroit.

Distorted Information

Reviews in the press of a report just released by the Federal Power Commission are likely to be very misleading to those not in close touch with present activity in the power field. One metropolitan daily quotes the report to the effect that "statistics reveal a marked trend in recent years toward construction of hydroelectric plants, this type now exceeding steam generation for the second year." It is difficult to discover the basis for such a statement and a perusal of the report will probably reveal qualifying information, not quoted in the press, that will give quite a different view of the situation.

According to the latest available figures of the Edison Electric Institute the installed steam capacity, among electric utilities, is considerably more than two and a half times the hydro capacity and the total annual output for a number of years past has averaged about sixty per cent generated by steam and forty per cent by water, there being slight variations from year to year due to droughts or flood conditions.

It is true that from 1931 up to the present year little new capacity, either steam or hydro, was added by the utilities and it is likely that the Federal Power Commission's report referred particularly to the Government's hydro projects under construction in 1935. One must not be mislead, however, by the publicity that has attached to these projects. While some are laid out for large ultimate capacities, to be installed if and when the load warrants, that installed initially, and available for producing power by the end of 1937, is greatly exceeded by the aggregate capacity of steam plant extensions now under construction or contract.

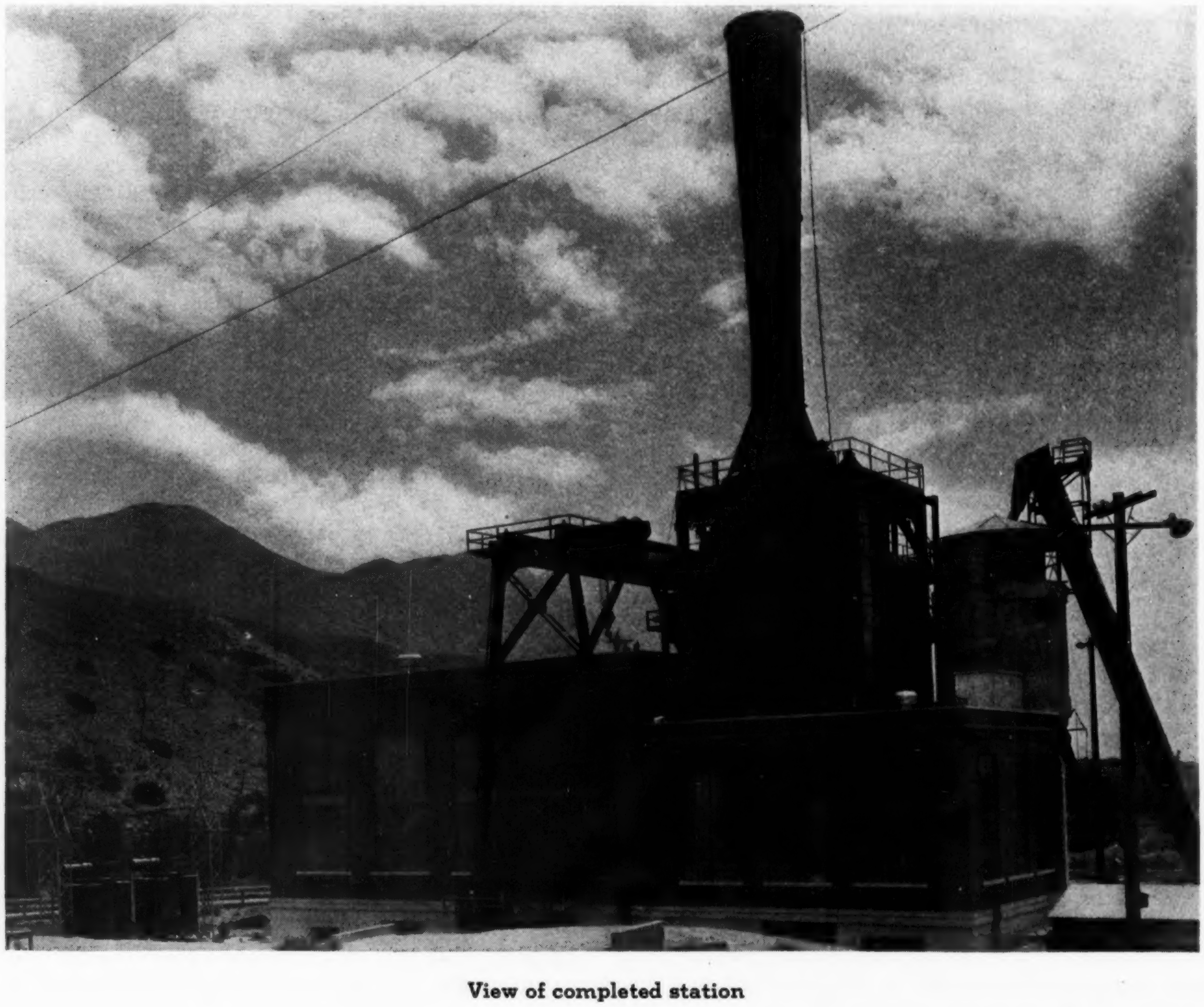
Too much misinformation is already current concerning power matters and it is most unfortunate that the uninformed are fed up with only partially digested facts.

SEMI-OUTDOOR STATION PLACED IN SERVICE

This 400-lb station of the Utah Power & Light Company is laid out on the unit plan and contains a single 200,000 lb per hr boiler supplying steam to an 18,750-kw turbine-generator. Only the lower part of the boiler is enclosed and the turbine room has unusually low headroom, thus effecting considerable saving in building cost. Because of the semi-outdoor construction special arrangements had to be made to protect the water columns, exhaust fan motor and other accessories, and to properly insulate the unit. The plant will supplement hydro plants on the company's system.

A NEW steam station embodying many unusual features of design was placed in operation this month by the Utah Power & Light Company. Located on the Provo River near Olmsted, Utah, about 50 miles from Salt Lake City and at an elevation of nearly 5000 ft, this plant of semi-outdoor construction will supplement the hydro power of the company's system. It is laid out on the single unit plan and consists of one boiler supplying steam at 400-lb pressure, 750 F temperature to an 18,750-kw General Electric condensing turbine-generator with 3-point bleed for feedwater heating.

By enclosing only that portion of the steam-generating unit that must be constantly accessible for operation and by making the turbine room only 17 ft from floor to bottom of roof steel, an appreciable saving was effected in building cost, despite the additional expense incurred in protecting the exposed parts from the elements.



View of completed station

As will be seen by reference to the illustrations, the entire steam-generating unit and its appurtenances are carried on four outside columns. The whole upper portion of the unit is exposed excepting the water columns which have separate enclosures. A water-proofed steel casing of overlapped construction and suitable insulation is provided for the setting. At the top of the steelwork there is a concrete slab which carries a 67-ft Thermix venturi-type stack (5 ft diameter at the throat and 11 ft at the top) and twin induced-draft fans of 103,000 cfm capacity each at 8.3-in. pressure and 655 F. Considerable study was given to the induced-draft fan drives because of their exposed position. As a result the motors are of the totally enclosed type. They draw ventilating air from beneath the concrete slab and are vented out through the top in a downward direction. Thus they will be able to function even though buried in snow. The structure is designed to withstand a wind velocity of 100 miles per hour.

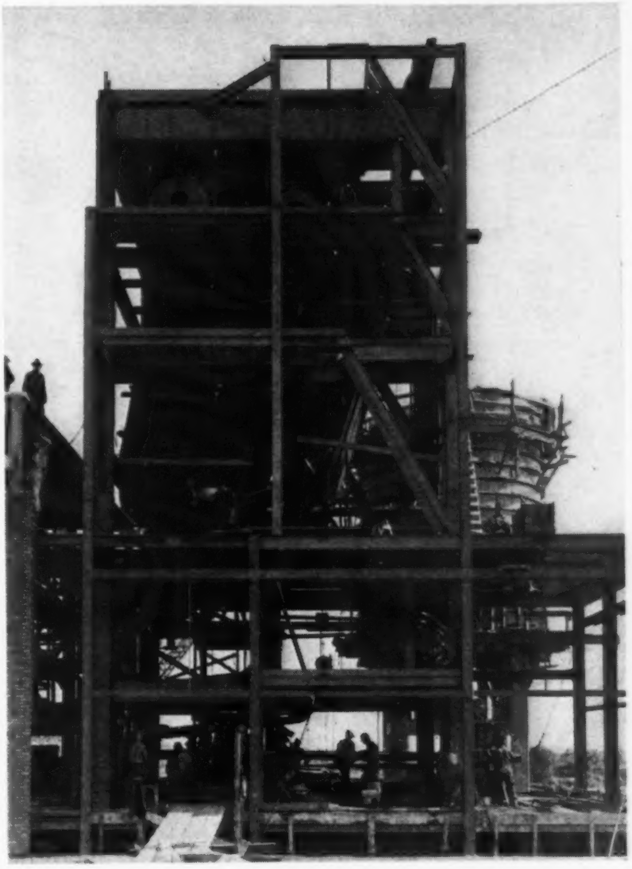
Protection for Water Columns

Special consideration had to be given to protection of the water columns and the Stets feedwater regulator. As there is no circulation in the piping from the drum to the water columns and Stets regulator, low-pressure heating pipes were run paralleling this piping within the insulation to guard against the possibility of freezing. For the same reason, a heated enclosure was provided around the feedwater regulator valves. Also, instrument piping containing water was buried in the insulation of nearby piping. The water columns—one a Reliance and the other a bi-colored Diamond—each register through a reflected light ray to the operating floor. Each is housed in a compartment, the underside of which connects with a 27-ft vertical duct leading to the boiler room roof. Each duct not only serves as a path for the light rays from the water columns but also encloses the drain pipes and conducts warm air from the operating floor enclosure to the water columns. Furthermore, the building enclosure is air-cooled to add to the comfort of the operators.

Features of Boiler and Setting

The boiler is of the CE-three-drum, bent-tube type having 18,600 sq ft of heating surface and rated at 200,000 lb of steam per hour, continuous. The front bank has 3-in. tubes and the middle and rear banks 2 in. with a row of 3-in. tubes in front of the middle bank separated by a baffle from the other tubes in this bank. It is designed for 450 lb per sq in. and will be operated at about 435 lb. An Elesco superheater of 3850 sq ft surface raises the total steam temperature to 760 F. The drums are all fusion welded, the top drums being 54 and 48 in. diameter and the bottom drum 48 in. A screen-type steam washer is installed in the larger of the top drums. Consolidated safety valves are provided as well as Edward blowoff, check and feed valves and Diamond soot blowers.

The setting includes combination front and rear water-cooled arches and water-cooled side walls employing plain tubes, also a water-cooled front wall. The front and rear arches are formed by placing T-tile between the tubes. This is backed with firebrick which, in turn, is backed by rockwool and the casing. The coal which averages 44 to 49 per cent fixed carbon, 40 to 41 per cent



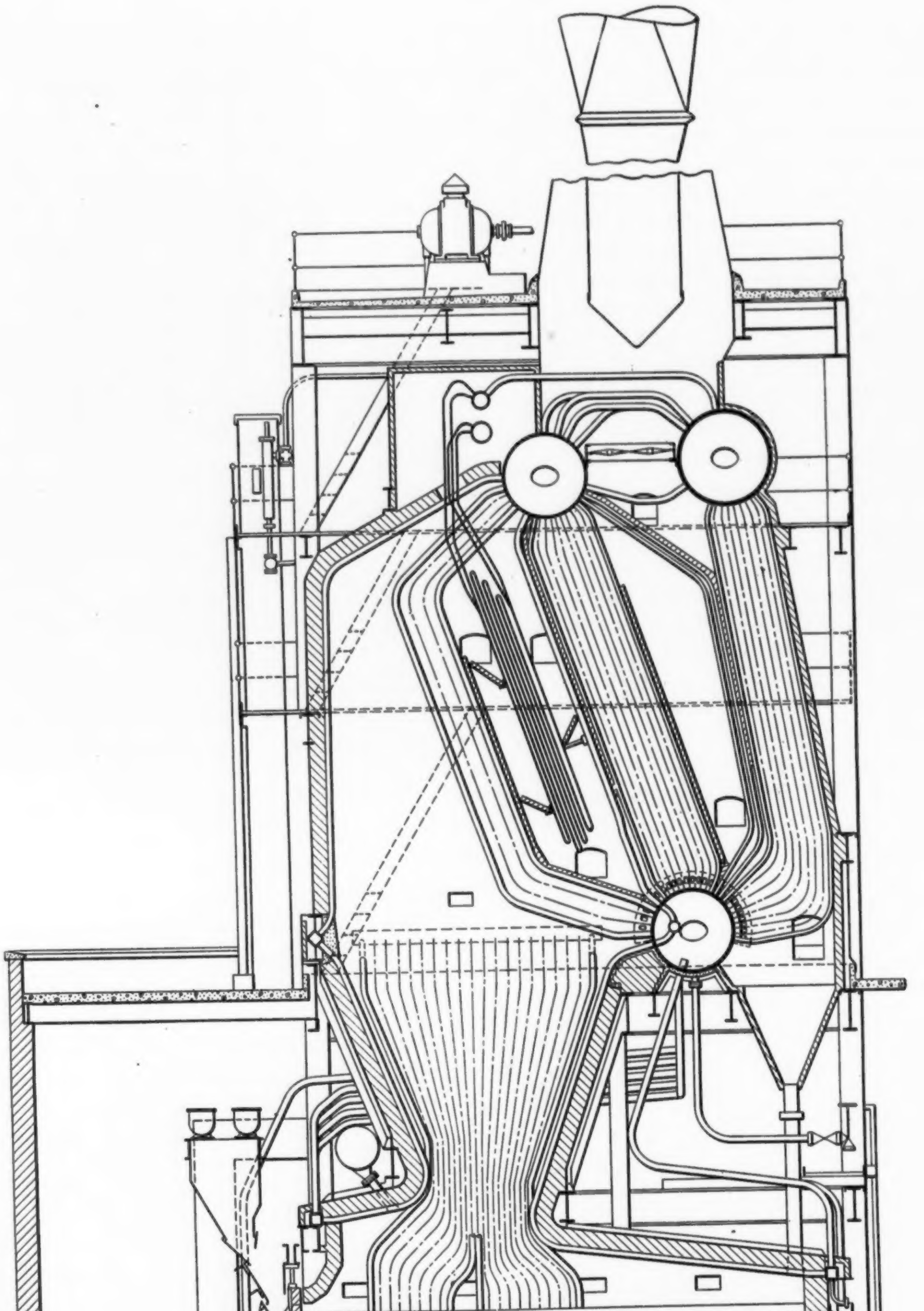
Construction view of boiler plant

volatile, 5 to 7.7 per cent moisture, 5.6 to 7.3 per cent ash, with an ash fusing temperature of 1880 to 1945 F and around 12,900 Btu per lb, has heretofore been regarded as an underfeed stoker fuel, because of its low ash content and its caking characteristics. The decision to employ a forced-draft chain-grate stoker with water-cooled front and rear arches was made only after it had been conclusively demonstrated that this could be done satisfactorily by burning fourteen carloads of this coal in another station similarly equipped. The stoker has an active length of 21 ft and width of 20 ft and is provided with side wall clinker chills.

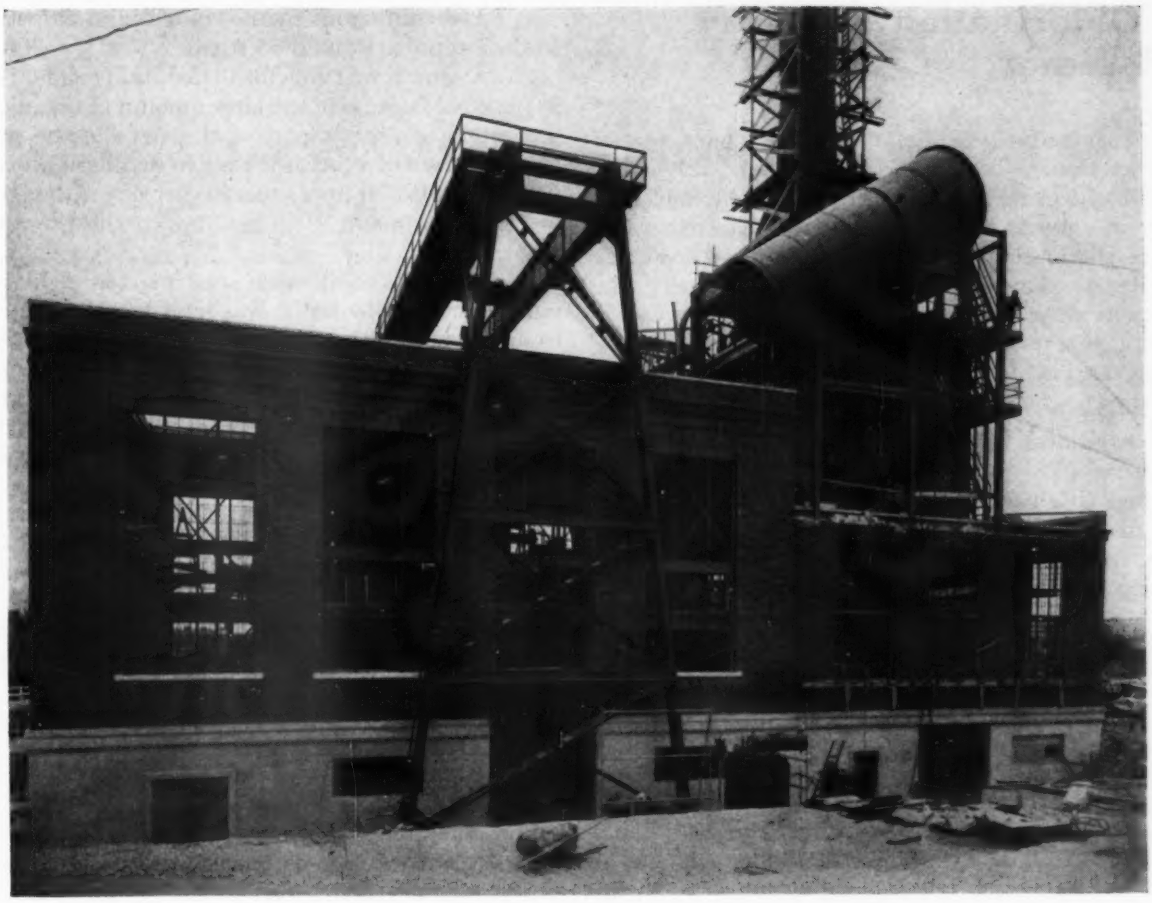
Over-fire air is supplied. This is taken from beneath the hood over the front of the stoker and is introduced through the lower part of the front arch by a 9000 cfm American Blower fan operating against 4.8 in. static pressure. The furnace volume is 5900 cu ft.

Forced draft is furnished by a 90,400 cfm motor-driven American Blower fan operating against 3.8 in. static pressure. In rating all the fans the fact that the station is at an elevation of 4840 ft above sea level had to be taken into consideration. A 485 cu ft ash hopper is placed at the rear of the stoker and fitted with self draining gates of the Allen-Sherman-Hoff type. Also under the front of the stoker there is installed a siftings hopper.

An outdoor coal storage field of 7500 tons capacity, with drag scraper equipment is located adjacent to the plant. Coal is elevated from a track hopper to a silo located beside the boiler room and two screw conveyors conduct the coal from the underside of the silo to the top of the stoker extension hopper. This hopper, be-



Section through boiler and setting



Construction view showing gantry crane over turbine room for lifting turbine-generator parts or auxiliaries through roof hatches

cause of its large size, is so arranged as to avoid packing of the coal and insure a continuous feed. Only one screw conveyor is required and its operation is controlled automatically by the level of the coal in the extension hopper. Means are provided to avoid segregation of fine and coarse coal at delivery to the silo and again at the extension hopper.

Outside Gantry Crane Provided

As previously mentioned, the headroom in the turbine room is low, only 17 ft, but a 25-ton outside gantry crane which extends over it is able to pick up any part of the turbine through hatches in the roof. In the turbine-room basement are located the condenser, heaters, pumps and other auxiliary equipment.

The station is located on the Provo River about a quarter of a mile below a hydro plant and condensing water is taken from an irrigation canal that is supplied from the tailrace of the hydro plant. The Provo River is also the source of service water and boiler feed make-up, which latter is evaporated after zeolite treatment.

Selection of a moderate steam pressure and temperature, simplicity of layout, the omission of heat-recovery equipment, semi-outdoor construction and single unit design were all factors contributing to minimum investment cost at the same time insuring a reasonably low station heat rate and maximum reliability. Credit for the design belongs to Ebasco Services Incorporated of New York, Consulting Engineers for the Utah Power & Light Company.

Figuring Government Power Costs

Writing on "Hidden Costs at Muscle Shoals" in the July Bulletin of the Edison Electric Institute, W. M. Carpenter, Economist of the Institute says:

"When a power company builds a dam, it usually has to borrow money to finance it. Bonds are sold and interest is paid as the individual coupons are presented. Until the plant begins operation, the interest which has accrued during construction is commonly added, along with other overhead items, to actual construction expenditures and constitutes an integral part of the investment upon which the project is expected to yield a return. After the plant goes into operation, interest must be provided out of the revenues of the enterprise.

"Regardless of the fact that bonds have been sold, the cost of a government enterprise, as such cost is reckoned by the Government, represents the actual out-of-pocket payments for the job. Interest on any specific project disappears into 'Interest on the Public Debt.'.....under the system of financing deficits by borrowing more money, additional bonds have to be sold to pay interest on money already borrowed: This results in still more interest. Government accounting ignores this very real element of cost."

Based on accepted methods of accounting Mr. Carpenter figures that for every kilowatt-hour sold by TVA in the 1935 fiscal year, the taxpayers of the United States donated a cent and a half.

The Chlorination of Spray Pond Water

In a paper before the recent A.S.M.E. Semi-Annual Meeting at Dallas, Texas, R. J. Crowdus of the Dallas Power & Light Company told of the experiences of that company in chlorination of condenser circulating water and algae elimination. While much of this experience was similar to that encountered by others during the development stage of chlorination treatment, the Dallas plant employs a spray pond and the effectiveness of chlorine in preventing algae growth in the pond and clogging of the spray nozzles warrants mention.

Previously much trouble had been experienced by what was believed to be carbonate scale in the spray nozzles and this had to be removed once a month by reaming out the nozzle throats. Investigation disclosed, however, that algae in the circulating water was of the form cystiphoral and sheathed itself in a siliceous envelope. As the algae passed through the condenser tubes it would attach itself to the sides of the tubes, collecting inorganic matter and forming slime. The algae in the slime would die and the slime would change to a hard scale.

Moreover, the algae would often collect at the bottom of the spray pond in large sheets which, when attaining a thickness of $^1/_4$ to $^1/_2$ in., would break loose and lodge in sheets on the intake screens. A sudden change in temperature during the fall and spring months would aggravate the tendency to break loose and obstruct passage through the screens.

In studying the algae it was found that a slight greenish substance was being pumped from the wells which furnished the greater part of the 20 to 30 million gallons of spray pond make-up each month. This greenish substance would easily pass through a sand filter. If left standing for a few days in a test tube it would settle to the bottom and grow rapidly. This proved to be seed algae and was one of the sources of the growth found in the spray pond.

Copper sulphate was at first used in the spray pond to inhibit the growth of algae but its effectiveness was reduced by the high sodium content of the circulating water which caused the copper sulphate to precipitate as a white powder. Chlorine was then tried and its use has effectively eliminated the growth in the spray pond, clogging of the nozzles and condenser tube accumulations.

Since chlorine gas is very soluble in water, it was at first decided to convey the dry gas into the pond through a needle valve and a rubber hose, but it was soon found that, although very soluble, it went into solution so slowly that the bubbles would either escape from the water or go into the circulating system before being dissolved with the possibility of damaging the condenser. Also, ordinary needle valves were not found satisfactory for controlling the flow. It was therefore decided to dissolve the gas completely in primary water and to inject this into the system. A high-grade air hose was found satisfactory for conveying this concentrated chlorine water.

Conclusions reached as to the best methods of injection and the benefits due to the use of chlorine gas are as follows:

1. The continuous method was found to be superior to the intermittent method tried.
2. At first it was difficult to maintain residual chlorine in the pond because of the large amount of organic matter but after the greater part of the algae in the pond had been disposed of it became easy to maintain a residual of 0.05 ppm with 50 lb of chlorine per day although this is increased to about 70 lb per day in the three summer months.
3. At first the chlorine was injected right into the pump suction pipe but it was later found more effective to admit it some distance out in the pond to allow the chlorine to act on the algae before entering the condenser. Now the chlorine is injected at three different points in the pond, each about 300 ft from the pump intake.
4. The chlorine did not attack the tubes nor any part of the condensing equipment.
5. In addition to treating the circulating water, the water in the settling tank is also treated to kill the seed algae and make cleaning of the tank unnecessary.
6. Savings and costs:

(*a*) Elimination of condenser tube deposits resulted in a decrease in terminal temperature difference of 3 deg F which represented a saving of $4000 per year.

(*b*) Reduction in the labor of cleaning tubes, spray nozzles, intake screens and settling tanks amounts to about $3000 per year.

(*c*) The cost of chlorine used is $1^1/_2$ cents per million gallons of water circulated, or about $1000 per year, but this may now be decreased as the algae have been killed.

(*d*) The net saving due to the use of chlorine treatment is about $6000 per year.

Morris E. Leeds, President of Leeds & Northrup Company, Philadelphia has been doubly honored by receiving the Henry L. Gantt Gold Medal awarded annually by the Institute of Management for "distinguished achievement in industrial management" and by the degree of Doctor of Engineering from the Polytechnic Institute of Brooklyn.

F. P. Fairchild has been appointed mechanical engineer of the Public Service Electric & Gas Company of New Jersey. Mr. Fairchild was for a number of years associated with United Engineers & Constructors of Philadelphia.

Ely C. Hutchinson, formerly president of Edge Moor Iron Company and later connected with the engineering department of J. G. White Engineering Corporation has recently become associated with Alco Products Company, New York.

George E. Seabury, who retired last year as superintendent of station engineering of the Edison Electric Illuminating Company of Boston, died on July 13th at New Haven, Conn. Mr. Seabury was widely known in power engineering circles and previous to his long connection with the Boston Edison Company had been identified with the construction division of the General Electric Company. He was 70 years old.

Some Recent Developments in Operation and Maintenance of Surface Condensers

By A. H. MOODY
Chief Chemist, New York Edison Co., Inc.

The several methods of maintaining clean condenser tubes are discussed and the advantages of hypochlorite solution are pointed out; practical suggestions are given for maintaining minimum air leakage, the improved performance of aluminum bronze alloy, containing 95 per cent copper and 5 per cent aluminum, or aluminum brass, containing 75 per cent copper, 22.5 per cent zinc and 2.5 per cent aluminum, as compared with admiralty metal for tubes is cited; and an improved method of removing condenser tubes is described.

MODERN use of large capacity steam turbine-generators has called for corresponding development in surface condenser effectiveness. Since the steam efficiency of a turbine depends upon the exhaust steam temperature which in turn is associated with condenser vacuum, improvement in condenser operation is a worthy object resulting in power station economy.

It can be shown that a gain of 0.1 in. in the vacuum, as measured by the mercury column, can reduce the steam consumption at an average load 0.5 to 0.6 per cent. A saving of this amount would be a welcome economy in any power plant and on a large station runs into several thousands of dollars per year. For instance, consider a 50,000-kw machine operating at an average load of 30,000 kw for 4000 hr per year. At a steam cost of 25 cents per 1000 lb a saving of one-half of one per cent is equal to $1500 annually produced by a tenth of an inch gain in vacuum.

Maintenance of condenser vacuum depends upon preserving the heat transfer rate between the circulating water and the steam space. Dirty condenser tubes, fouled by slime and scale, are a cause of serious loss in condenser efficiency. Certain slime conditions are effectively prevented by hourly injections of chlorine[1] or chloramines[2] into the circulating water. These methods in some cases have eliminated the necessity of plug cleaning for a number of years.

Chloramine solutions made by dissolving equal parts of bleaching powder and ammonium sulphate have been recommended where the chlorine demand of the water is high as shown by the orthotolidene test.[3] Sodium or calcium hypochlorite solutions may be made by bubbling chlorine gas into a cold solution of lye or lime but this preparation requires some care to prevent escape of the obnoxious chlorine gas. Attempts have been made to treat the empty but wet condenser with the chlorine gas directly but such efforts have generally resulted in serious localized corrosion within the condenser. Potassium permanganate was found to be of no value.

Another effective method of removing slime and scale from condensers consists in drying out the tubes for 24 hr by passing hot air at 120 F, from the generator air ducts into the condenser by means of a canvas "blimp" about 14 in. in diameter, and at the same time 140 F steam is slowly admitted to the steam space and vented through opened relief valves. When the scale is thoroughly dry it curls loose from the tubes and may be washed out of each tube by a high-pressure water jet. This method used about every three weeks has entirely eliminated the necessity of plug cleaning. Opening an idle condenser when the unit is out of service is good practice to allow the scale to dry out by normal air circulation. This frequently gives improved vacuum at no cost whatsoever.

Air leakage is another cause of lost vacuum and consequently higher exhaust steam temperature for a given load and inlet water temperature. The "bell jar" tests performed by operators at regular and frequent intervals are of some value in following up total air leakage into the condenser and thereby initiating an investigation when the leakage becomes excessive. But the total air leakage is not a measure of its effect on vacuum since the latter depends upon the point of leakage as well as the total quantity of air. Since both load changes and inlet water temperature changes affect the vacuum, these factors must be measured accurately and corrected before an evaluation of the loss due to air leakage or dirty condenser tubes may be made.

Fig. 1 is a vacuum-correction chart to reduce vacuum to 70 F inlet circulating water temperature and 30-in. barometer. In the example given the inlet-water temperature of 55 F is found under a turbine load of 35,000 kw and projected horizontally to the right-hand reference line. From this point a straight edge connects the observed absolute pressure 0.70 in. of mercury, corresponding to the exhaust temperature of the turbine.

[1] U. S. Pat. No. 1,745,141.
[2] U. S. Pat. No. 1,989,380.
[3] Amer. Pub. Health Assoc. Methods. p. 44 (1933).

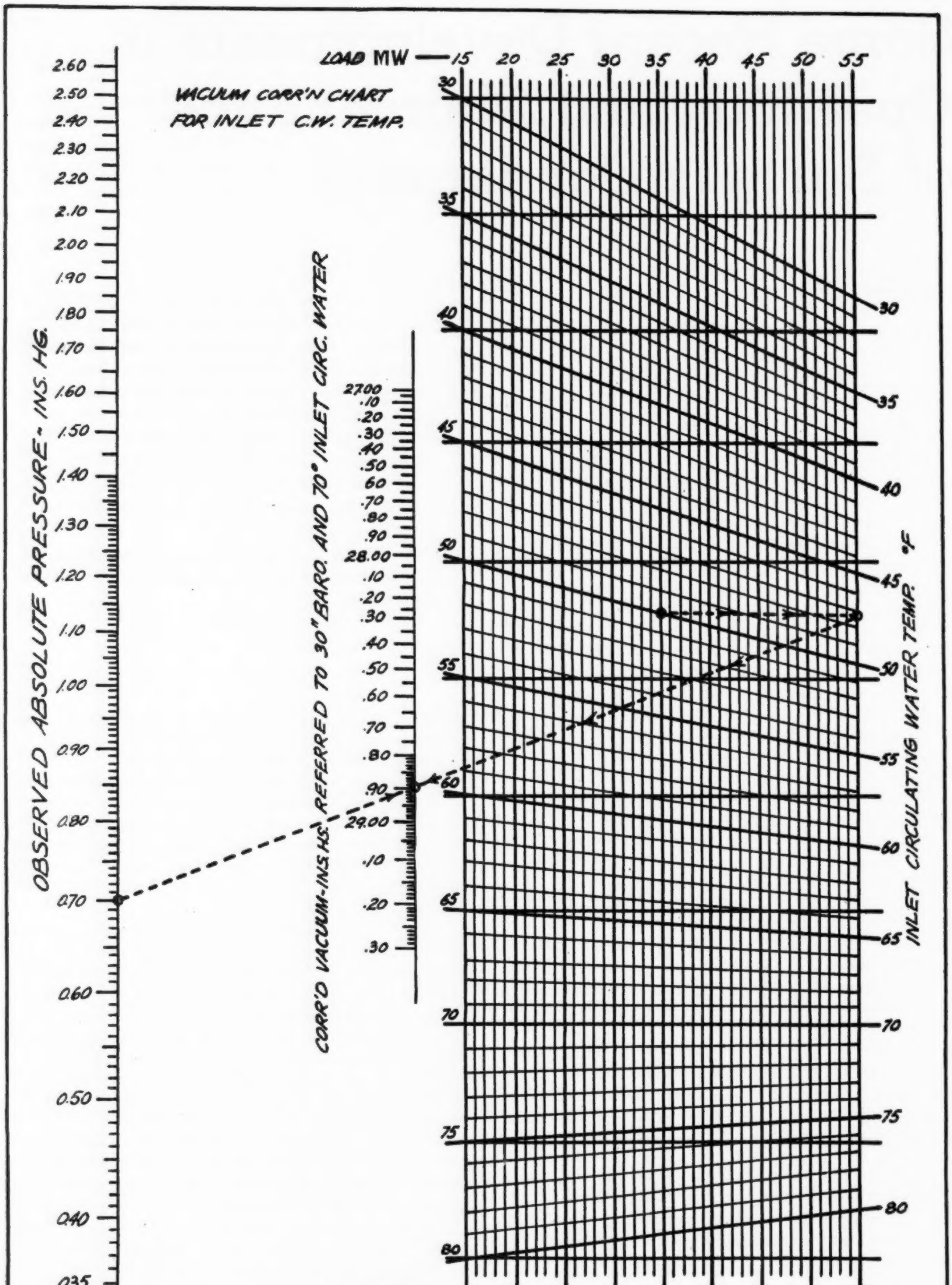


Fig. 1—Vacuum correction chart for inlet circulating water temperature

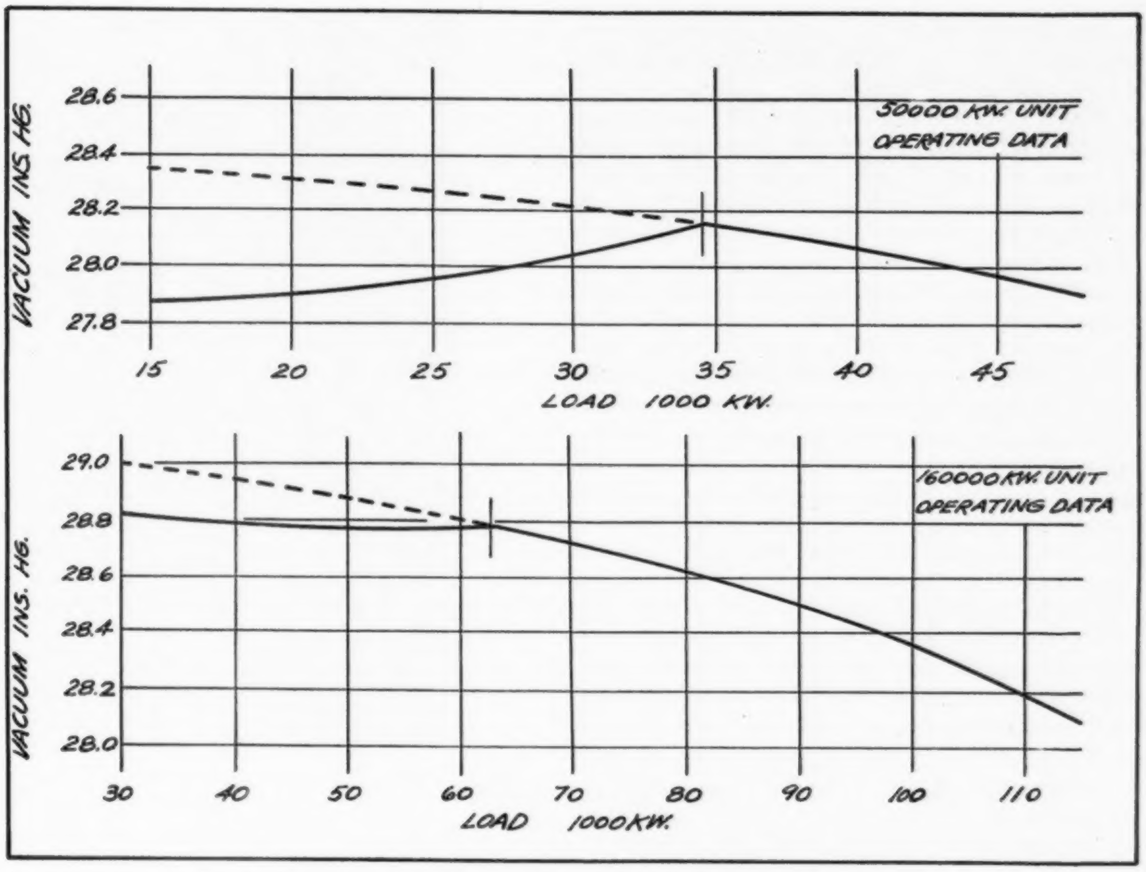


Fig. 2—Vacuum-load curves for 50,000-and 160,000- kw units

The corrected vacuum is read directly as 28.9 in. Hg on the middle scale. When the corrected vacuum is plotted against load from hourly log readings a curve may be obtained similar to those shown in Fig. 2.

The curves in Fig. 2 have been selected to illustrate certain features. In both curves there is a sharp break at a definite load. A study of the design data of these machines, both of which are of the cross-compound type, shows that the break in the curve occurs at a load at which the pressure in the receiver pipe between the high- and low-pressure sections is exactly atmospheric. At greater loads the pressure is higher so that no air leakage occurs and the vacuum curve is normal. A mild break of this sort is shown when the high-pressure gland seal becomes imperfect, while a sharp drop is indicative of air leakage between the two sections. A leak at the condenser would affect the entire curve adversely. By determining the point of leakage in this manner attention can be directed more closely to the seat of the trouble, thereby improving a difficult task. A study of any given machine and its operating data makes it possible to predict the general location of leaks from their effect on the load-curve characteristic. Application of this method of analysis to daily operating data serves to detect maintenance failures before they become serious, thereby producing an overall improvement in efficiency.

Condenser-tube failures due to corrosion or inlet-end erosion are the cause of expensive maintenance, loss of turbine efficiency, boiler feedwater contamination and long turbine outage for retubing.

Special alloys have been recommended and several have proven to have much longer life than even admiralty

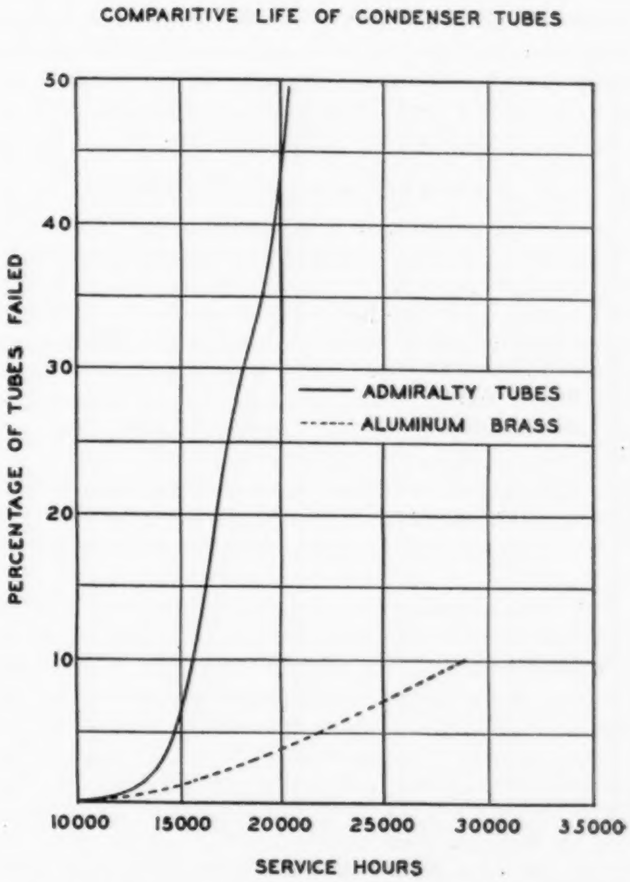


Fig. 3—Comparative life of admiralty and aluminum brass condenser tubes

tubes in the same condensers. An aluminum bronze alloy containing 95 per cent copper and 5 per cent aluminum has been used for 23 years in one condenser with a record of 88,000 service hours while a similar condenser in the same station shows a life of only a few years using admiralty tubes. A lighter and cheaper alloy of aluminum brass, containing copper 75 per cent, zinc 22.5 and aluminum 2.5 per cent, has proven superior to admiralty metal. Fig. 3 shows the comparative life in service hours of admiralty and aluminum brass alloys. Each curve represents 16,000 condenser tubes in the same condenser, the aluminum brass being used to replace the admiralty. The aluminum brass shows only 10 per cent failures in 28,000 service hours while the admiralty tubes had to be completely replaced at the end of 21,000 hr after more than 50 per cent failures. Development of the extrusion method of making alloy tubes from a solid casting and the use of underpouring in the casting process have improved the quality and reduced the cost of these tubes.

Inlet end failures due to turbulent flow have been attacked by several methods. Since the corrosion occurs for a distance of only six to eight inches, protection has been successfully afforded by butt-welding a one-foot length of corrosion-resistant tube to an ordinary admiralty condenser tube. Monel metal, aluminum brass and aluminum bronze have been so applied. In one case tubes were purchased longer than required and the extra length allowed to project into the discharge water box. When inlet end failures occurred the tube was drawn back and the inlet end cut off, thus effecting nearly double life from the tubes. In one condenser 1300 tubes had been plugged mainly because of inlet end failures. These tubes were painted with a special cement for about eight inches into the tube. More than 800 tubes responded to the treatment and are still in service after several years. In other cases two or more coats were necessary before a satisfactory sealing was obtained.

Cutting and Removing Old Tubes

Considerable saving in time and labor involved in retubing condensers has been made by recent developments in the technique of removing and cutting old tubes and in packing. The methods also preserve the condition of the old tubes, allowing partial recovery or higher scrap prices where the tubes have only localized defects. Where the tubes are rolled at both ends one end may be cut off by means of a newly designed rotary cutter inserted into the tube. This cutter is electric or air driven and cuts each tube at any predetermined point. The cutting of the tube requires only a few seconds so that a single operator can cut hundreds of tubes in a relatively short time.

In pulling old tubes the tube does not draw easily until the packing is loose. This may not happen until the tube has been drawn out ten or fifteen feet. A mechanical tube puller has been introduced which loosens the tube and packing simultaneously with a tube travel of less than one foot. Four concentric teeth on the puller internally engages the tube (and packing if desired), withdraws the tube a few inches, and disengages the drawing tool in a single operation. A 100-lb compressed air supply furnishes sufficient power to withdraw the tube. One or two operators are required to handle the machine and under favorable conditions can loosen 15 to 25 tubes per minute. Once the tube and packing are free a wet tube may be easily drawn through the support plates and tube sheet by a single workman. Due to complete freeing of the tube there is little damage to the tube in the pulling operation. In some cases where tube failures were confined to eight inches on the inlet end a short portion was cut off and the tube used immediately in a shorter condenser. Fig. 4 shows the tube puller in position to withdraw a tube on a 10,000-tube condenser where isolated failures were to be replaced by new tubes.

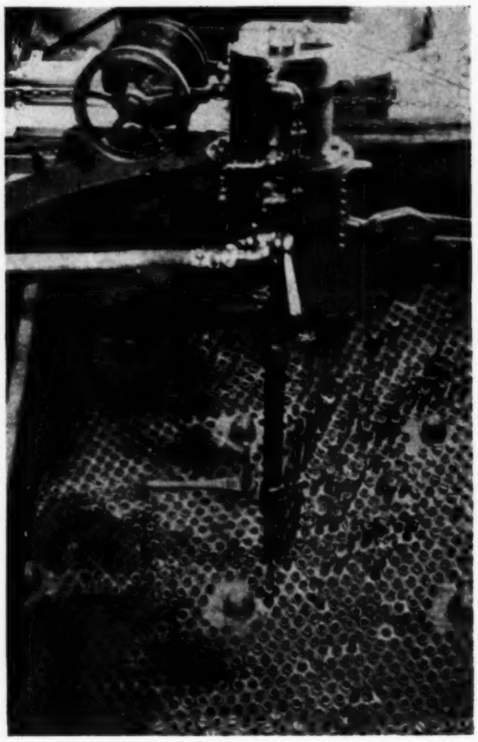


Fig. 4—Removal of condenser tubes by automatic tube puller

Improved methods of keeping condenser tubes free from slime by chemical treatment of the condenser, drying out condenser tubes when idle, and following up air leakage conditions by corrected vacuum-load curves are effective means of maintaining turbine efficiencies. Maintenance costs of condensers have been reduced by the use of new alloy tubes and improved technique in tube removal.

Delray to Burn Blast Furnace Gas

The Great Lakes Steel Corporation has awarded a contract to the Semet-Solvay Engineering Corporation, of New York, for the building of a pipeline which will convey blast furnace gas from the Hanna furnaces, on Zug Island, to the Delray plant of the Detroit Edison Company. The gas line will be 48 in. in diameter and will be of welded steel construction. It will start at the Hanna Furnace plant and cross the Rouge River over a new bridge, to be constructed, and will then run some 2400 ft to the Delray plant.

The river crossing will be approximately 200 ft and will clear the high water line by 150 ft.

Rating and Performance of Stationary Steam Boilers

By W. S. PATTERSON

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The author discusses the absurdity of the terms "boiler horsepower" and "per cent of rating" as applied to modern steam generating units and suggests more rational expressions. He analyzes the effect of variable entering gas temperature, operating pressure and superheat, on the boiler performance, illustrating these by curves and equations. Gas temperature gradient and draft loss through boilers are also discussed.

Boiler Rating

History

When boilers were fed by gravity through vertical feed pipes extending only 7 ft above the water level, the "Directions" prescribed that a pressure sufficient to support a column of mercury 1 to 2 in. high should be sustained. These "Directions" were published in 1779 by Boulton and Watt, entitled "Directions of Erecting and Working of the Newly-Invented Steam Engines." They stated further that higher pressures "do not serve any good purpose."[1] Many of these boilers proved too small for the engines they were intended to serve but for some time Watt used a standard of 8 sq ft of heating surface per cu ft of water evaporated per hour. In terms of "equivalent evaporation" that would be approximately 7.5 lb per hr equivalent evaporation per square foot of heating surface. It is probable that, even in those days, the method of rating boilers and stating their expected capacity caused less confusion and was more logical than one of the methods commonly used in the industry today, i.e., stating size in "boiler horsepower" and capacity in "per cent of rating."

In the days when the steam boiler was an adjunct to the steam engine it was convenient to rate the former in the same units as the latter. Consequently, when, as the result of tests in connection with the Centennial Exposition in 1876, it had been determined that a good engine required approximately 30 lb of steam per hour to produce a horsepower it was logical for the Centennial Committee to recommend that a "boiler horsepower" be defined as the evaporation of 30 lb of water per hour at a temperature of 100 F into steam at 70-lb gage pressure. This definition was standardized by the A.S.M.E. in 1889 but later converted to "equivalent evaporation" in the A.S.M.E. Code of 1915, when a boiler horsepower was defined as 34.5 lb per hr equivalent evaporation from and at 212 F. Although engines are no longer the principal users of steam from boilers and engine efficiencies have been greatly improved, the old definition has been tenaciously adhered to and is still commonly employed for the calculation of developed boiler horsepower.

The mere act of defining "boiler horsepower" did not, however, solve the problem of designing a boiler so that it would deliver what was expected of it. A given size of boiler has a steaming or "horsepower" capacity which is quite variable, depending on draft available, and many other factors pertaining to the furnace, the fuel, arrangement of baffles, etc. Many of these factors were not under the control of the boiler manufacturer, but it was determined by test that if 10 or 12 sq ft of surface were used as the basis of each "boiler horsepower," then the capacity so calculated from the installed surface could be obtained quite comfortably and with better efficiency than if the same capacity were obtained with less boiler surface by forcing the fires.

In this manner the surface per horsepower was decided upon and practically all the boilers reported by Barrus in his rather extensive book of early boiler tests[2] had their ratings based on 12 sq ft per boiler horsepower. Manufacturers later standardized on 10 sq ft per rated boiler horsepower, and although some small boilers are still manufactured which can only comfortably deliver 34.5 lb per hr equivalent evaporation per 10 sq ft of surface, the economical capacity of the average large unit so far exceeds this figure that engineers have for many years recognized the need for a new and entirely different method of rating boilers.

Present Methods of Rating Boilers

For a number of years many boilers have been guaranteed by their manufacturers to deliver three to five times more steam than their rated capacity. The expression "per cent of rating" was adopted as a means of stating this ability to produce more than would be expected by the old-time standards. The expression has real significance so long as it applies only to a conventional boiler, but when the heat absorbed by an economizer and superheater are added to that absorbed

[1] See Combustion, January 1936, "James Watt, Engineer and Scientist."

[2] "Boiler Tests," by G. H. Barrus, 1891.

by the boiler and the total divided by 970.2 to determine the "equivalent evaporation," then the calculation of "per cent of rating," based on boiler surface only, becomes quite meaningless. This is the procedure that has been followed for about twenty-five years, during which time the extensive use of water-cooled furnaces connected into the boiler circulation has made the term "per cent of rating" still more meaningless. Now we are in the era of completely water-cooled furnaces, high operating pressure and high total steam temperature and the time has come to reconsider what constitutes a boiler, because it is difficult to find more than the mere skeleton of the conventional boiler upon examination of the drawings of some of the most recently proposed units.

The confusion which results from the careless use of the word "boiler" may be illustrated by reference to Fig. 1. In this steam generating unit what constitutes the boiler? The only recognizable components of a conventional boiler are the steam drum, the short vertical headers, the tubes between the headers and the circulators connecting the headers to the drum. The manufacturer has, therefore, in accordance with the custom of many years, considered only those parts enumerated above as the "boiler." But where in the gas flow path is the boiler outlet? So far as heat absorption by the "boiler" is concerned, the boiler outlet might be considered as located at the superheater inlet, because beyond that point those portions of the boiler circulators which are not refractory covered, or otherwise removed from contact with the gases, are not very effective. The side wall tubes in the superheater zone are connected into the water wall circulation and their surface therefore not included in the "boiler." The horizontal tube

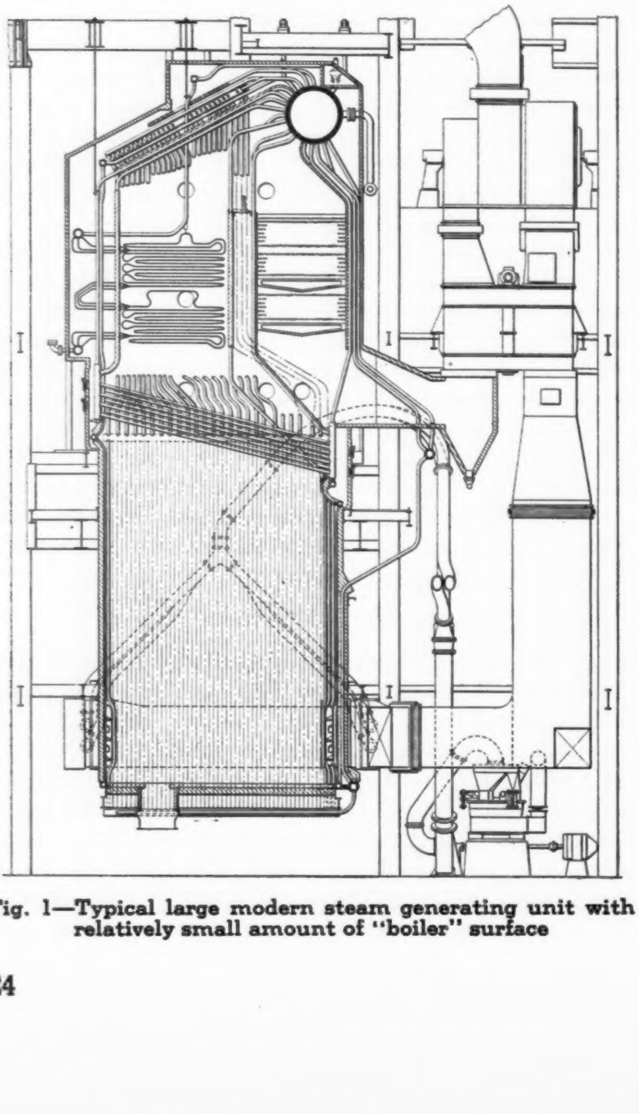

Fig. 1—Typical large modern steam generating unit with relatively small amount of "boiler" surface

banks below the steam drum, indicated in Fig. 1, are economizer surface. The gases pass first over the straight tube boiler surface, then the superheater, and then the economizer, and yet technically the "boiler" outlet might be considered to be beyond the economizer outlet because one of the boiler circulators is passed over by the gases after they have left the economizer. This particular row of tubes will actually reheat the gases at certain loads and yet according to existing rules for figuring surface the portion of its length exposed to the gases beyond the economizer could be included as boiler heating surface.

The "boiler" surface located below the superheater in Fig. 1 slightly exceeds 3000 sq ft and the exposed surface of the circulators and drum brings the total for the "boiler" up to 7000 sq ft. The maximum output of the unit is approximately 550 million Btu per hour so that the "per cent of rating" calculated in the usual manner would exceed 2300 per cent, developed by boiler, superheater, water-walls and economizer. That this method of expressing capacity is meaningless is quite obvious. Contrary to what might be expected, it does not indicate a high heat absorption by the "boiler," high steam velocity through the boiler tubes, high gas velocity through the boiler, high overall draft requirements, nor a great sacrifice in efficiency due to operation so far in excess of the so-called "nominal boiler rating" of 700 hp. This is because the water-walls, superheater and economizer absorb most of the heat but their surfaces are not included in the nominal surface of the boiler.

This steam generating unit is not, however, rated in nominal or expected boiler horsepower, nor is its guaranteed capacity expressed in per cent of rating. Use of these terms in connection with large units has long since been discarded by most manufacturers, although still used in connection with smaller capacity units, largely because of popular demand by the users of boilers, or because consulting engineers have written the specifications in these terms. This unit, including all the abovementioned components and the air preheater, is, like many others, guaranteed to evaporate a definite weight of water at a specified temperature into steam at specified pressure and temperature. This guarantee assumes that sufficient coal will be burned to provide the heat and sufficient draft provided to remove the products of combustion. With less fuel burned or less draft provided it would generate less steam. This steam generating unit has no "nominal" capacity and its maximum capacity may be limited also by other factors than the above.

Suggested Rating of Steam Generating Units

One manufacturer has for years used the term "steam generator" as representing a particular combination of boiler, furnace, etc. In stating the nominal size of these units the surface given is that of the convection and radiant heat absorbing surface except superheater, economizer and air heater. In an A. S. M. E. paper presented several years ago[3] the authors recommended "steam generator" to include everything from coal chute to breeching connection. They also recommended that the word "boiler" be dropped from future usage or restricted to refer to the shell only. The writer agrees that much confusion in the future would be

[3] "A Suggestion for Rating Steam Boilers," 1928, Shoudy and Jacobi.

eliminated if the word "boiler" were dropped entirely, but feels the term "steam generator" should be applied only to that group of component parts between the feed pipe and superheated steam outlet which handle water and/or steam. "Steam generating unit" could then be used to include air preheater, fuel burning equipment, draft equipment, etc., or all apparatus integral with or serving the one "steam generator."

The term "steam generator" should be used for the group of component parts mentioned regardless of whether they are furnished by a single manufacturer. Distinct sectionalizing of the parts by name, surface and heat absorbing capacity will, however, always be necessary because of this possibility of different manufacturers. The "steam generator" may, therefore, consist of:

Heater section
Steamer section
Water-wall section
Superheater section
Reheater section

Other appropriate names for the various sections might be selected. The water-wall section of some units evaporates more steam than the steamer section (now commonly called the "boiler") and technically these two should not have names indicating different functional characteristics because they both evaporate water into steam and it will be impractical to segregate their heat absorption capacities. Their surfaces must, however, be listed separately because there are many types and arrangements of water-wall tubes and blocks and as many ways of evaluating their effective heating surface, and because the same manufacturer may not supply both.

It is desirable to state the size of a steam generator in a way that will give a true impression of its physical size. The use of the combined surface of the component sections as the method of rating its physical size would, therefore, be preferable to the present habit of emphasizing only the size or horsepower of the so-called "boiler."

It is also desirable that the capacity of a steam generator be given in terms having thermodynamic significance. The difficulty here is that the heating surface of the unit is no more an indication of its capacity than the size of an automobile chassis, *sans* motor, is an indication of its speed. With a good fuel and a powerful motor, almost any reasonable speed can be obtained from the automobile. With adequate furnace, burning and draft equipment, and with good boiler water and ample drum capacity, a very high steam capacity can be obtained from a steam generator per square foot of installed surface. But, as discussed in a previous issue of this magazine,[4] each type of boiler has its limiting capacity factors which are independent of fuel burning capacity and available draft. There is, therefore, no way of arbitrarily rating steam generator capacity per square foot of combined surface that would apply to all types. It remains for the designer to calculate the maximum capacity from known operating conditions. The units in which this is stated should be such that the various components of the "steam generator" as previously defined may be rated separately in the same units. Obviously, therefore, "actual evaporation" or "equivalent evaporation" as now used are not entirely satisfactory, since literally the same weight of fluid passes through the economizer and superheater sections, although the surfaces and heat-absorbing capacity of these two sections may be different. A more satisfactory system would, therefore, be to rate the steam generator in "Million Btu per Hour." Each component could then be rated in the same units and their sum would equal the rating of the entire steam generator calculated by subtracting the heat content of the feed from the heat content of the steam in its final condition and correcting where necessary for reheat. The term would not be awkward, and the same figure would represent approximately the evaporation rate in 1000 lb per hr; or, divided by the fractional overall efficiency, this same figure would represent approximately the gas weight in 1000 lb per hr.

[4] "Boiler Capacity Has Outgrown Its Terminology," COMBUSTION, March 1934.

Boiler Performance

The Fundamental Equation

Boiler performance, as distinguished from capacity, refers to the efficiency of heat absorption and the draft required to overcome the resistance of the gases in passing over the surface. For many years it has been the custom to consider variations in draft loss and temperature performance as a function of output only, expressed in per cent of rating. But the performance of a boiler is more nearly a function of input, as will be shown by development of the fundamental equation for performance.

In the conventional type of steam boiler the feedwater is delivered to the steam drum and allowed to heat up to the temperature of saturated steam at drum-operating pressure before entering the boiler tubes. Heat transfer between the hot gases and the cooler liquid therefore takes place in a manner as illustrated by Fig. 2 where S represents the surface passed over, t_1, the entering gas temperature, t_2, the leaving gas temperature and ts the temperature of the saturated water and steam mixture.

It is obvious that the gases cannot be cooled to a temperature lower than ts and that this temperature can be reached only by passing the gases over an infinite amount of surface or by leaving the gases in contact with a lesser amount of surface for an infinite length of time. The amount of heat absorbed by the surface in a given length of time is equal to the product of the surface times the heat transfer per unit of surface in the same unit of time. The heat transfer "rate" is, by definition, generally expressed in heat units (Btu) per unit of surface (sq ft) per unit of time (hour) per unit of temperature difference (deg F) between the heat-absorbing fluid and the hot gas. Since the reduction in temperature between t_1 and t_2 is not directly proportional to the surface passed over, the application of the transfer

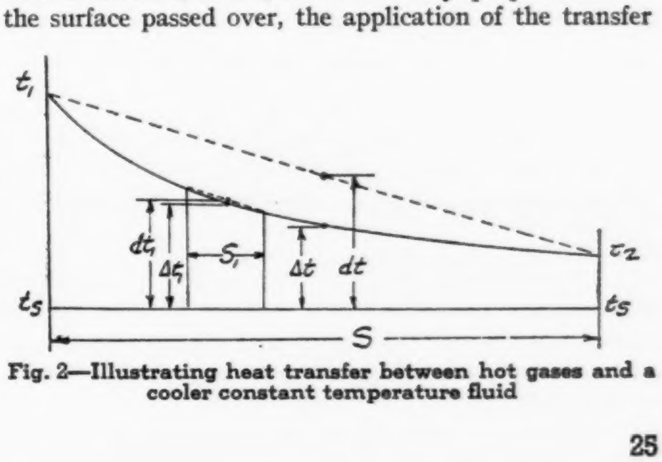

Fig. 2—Illustrating heat transfer between hot gases and a cooler constant temperature fluid

rate requires the use of the logarithmic mean temperature difference Δt (see Fig. 2), when a large amount of surface with considerable gas temperature drop is involved. It is obvious, however, from Fig. 2 that if a small portion of the surface such as S_1 is considered there may be very little difference between the logarithmic mean temperature difference Δt_1 and the arithmetic mean difference, dt_1.

Equation 1 expresses the relation between total heat absorbed, surface, heat transfer rate and temperature difference

$$H = S \times R \times \Delta t \quad \ldots\ldots [1]$$

where H = Btu per hour absorbed by surface S
S = surface in sq ft
R = heat transfer rate, Btu per sq ft per hour per deg F
Δt = logarithmic mean temperature difference between the two fluids

For a closed system the heat, H, absorbed through the surface S would also be equal to the heat given up by the gases in passing over it, which may be expressed as follows:

$$H = W_g \times C_g \times (t_1 - t_2) \quad \ldots\ldots (2)$$

where W_g = weight of gas in pounds per hour entering the system
C_g = mean specific heat at constant pressure between t_1 and t_2
$(t_1 - t_2)$ = gas temperature drop illustrated by Fig. 2

In practice, however, the gases are under a negative pressure with respect to surrounding atmosphere and it is impractical to attempt to prevent all infiltration into the gas stream. Consequently, since the infiltration air mixes with the gas stream and is discharged from the boiler at temperature t_2, it absorbs heat from the gas stream, and the heat quantity H, which is transmitted through the surface, S, to the water in the boiler may be expressed as follows for an actual boiler:

$$H = [W_g \times C_g \times (t_1 - t_2)] - [W_l \times C_l \times (t_2 - t_0)] \quad \ldots (3)$$

where W_l = weight of air leakage, lb per hr, entering gas stream
C_l = mean specific heat of leakage air between t_2 and t_0
t_0 = temperature of leakage air before entering the gas stream

By assuming average values for W_l and C_l based respectively on the expected CO_2 drop between furnace and boiler outlet and an assumed average temperature, t_2, Equation (3) may be simplified without much error and expressed in the form:

$$H = K \times W_g \times C_g \times (t_1 - t_2) \quad \ldots\ldots (4)$$

where K is a constant less than unity to allow for heat lost from the gas stream due to infiltration and also, if desired, to take care of radiation loss through the boiler setting walls.

Combining Equations (1) and (4) we have:

$$S \times R \times \Delta t = K \times W_g \times C_g \times (t_1 - t_2) \quad \ldots\ldots (5)$$

But it may be proved mathematically that:

$$\Delta t = \frac{t_1 - t_2}{\log_e \left[\frac{t_1 - t_s}{t_2 - t_s}\right]} \quad \ldots\ldots (6)$$

and combining (5) and (6), simplifying and transposing, we have

$$\frac{S \times R}{K \times W_g \times C_g} = \log_e \left[\frac{t_1 - t_s}{t_2 - t_s}\right]$$

or:

$$t_2 - t_s = \frac{(t_1 - t_s)}{\text{antilog}_e \left(\frac{S \times R}{K \times W_g \times C_g}\right)} \quad \ldots\ldots (7)$$

Equation (7) is in a form which may readily be used to solve for the temperature difference between saturated steam and gases leaving the boiler, or any part of the boiler, by proper substitution of known or calculated quantities in the right-hand side. Several possibilities of introducing considerable error are not apparent, however, from a general inspection of the equation. A word of caution is therefore deemed advisable. In the first place the derivation assumes a constant temperature of heat-receiving fluid and no variation in heat transfer rate on the steam and water side of the metal heat-transmitting surface regardless of rate of evaporation in the boiler or location of the surface.

In the second place the very definite connection between S, R and W_g must not be entirely ignored if extreme accuracy is to be expected. S is the effective heating surface for which the transfer rate R is applicable. R does, and S may, vary with different values of W_g because at low gas velocities part of the installed tube surface may be unavoidably bypassed and thus made less effective than at higher gas velocities. The test and calculation methods used in determining the values of R to be used in Equation (7) must, therefore, be understood and this is generally the reason why heat-transfer rates obtained by research laboratory methods and reported in textbooks cannot be applied directly to the full installed tube surface of a boiler without some possibility of error.

In the third place, going back to Equation (2), where the term W_g was first introduced it will be noticed that this is the weight of gas per hour entering the boiler. This gas weight should not be used in Equation (7), however, unless it is known that R is the true rate of heat transfer (i.e., the apparent rate corrected for the effect of infiltration into, and external heat loss from the setting). In working up most heat transfer rate data from boiler tests, only the gas weight leaving the boiler is calculated and the entire heat loss represented by the gas temperature drop between furnace and boiler outlet (or across any part of the boiler when the rate of heat transfer in a given pass or bank is being determined) is generally assumed to have been absorbed by the heating surface passed over. The transfer rate thus determined is obviously not the actual true rate and therefore, when using R determined in this manner in the application of Equation (7), the weight of gas used must be that leaving the boiler and K must be taken as 1.0. S may ordinarily be taken as 100 per cent of the installed tube surface passed over unless it is obviously partly ineffective due

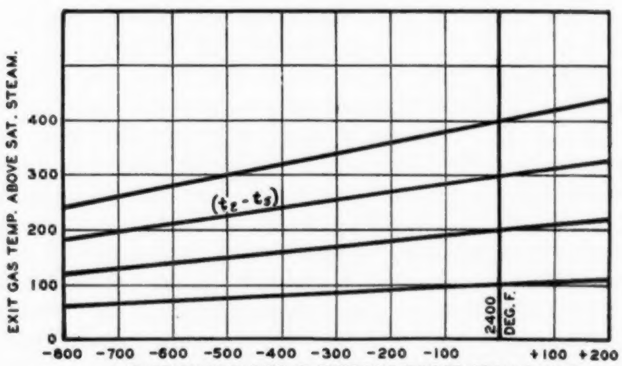

Fig. 3—Effect of variable gas temperature entering boiler on exit gas temperature

to location of baffles or other conditions which must be taken into consideration and an experience factor applied.

Effect of Variable Entering Gas Temperature

An examination of Equation (7) will disclose that, other things being equal, such as pressure, heating surface and gas weight, if $(t_2 - t_s)$ is assumed to represent the temperature above saturated steam of the gases leaving the boiler, the magnitude of the effect of changes in the entering temperature t_1 will depend on the magnitude of $(t_2 - t_s)$.

To illustrate how Equation (7) may be used to calculate the effect of variable furnace outlet temperature, assume that on substitution of $t_1 = 2400$ F, $t_s = 450$ F, and appropriate values for R, W_g and C_g, a certain boiler of surface S will reduce the gas temperature to within 200 deg F of saturated steam temperature ($t_2 - t_s = 200$ deg F). Then if t_1 be reduced to 2000 F, $t_2 - t_s$ will become $(2000 - 450)/(2400 - 450) \times 200 = 159$ deg F indicating a reduction of 41 deg in boiler outlet gas temperature for 400 deg reduction in entering gas temperature. However, had the boiler been capable of reducing the gas temperature from 2400 F to 100 deg above saturated steam, then reducing the entering temperature 400 deg would have had an effect of only 20.5 deg on the leaving temperature. The above calculation is based on $(S \times R)/(W_g \times C_g)$ remaining constant for changes in t_1 within the limits used above and plotted on Fig. 3. Actually R, W_g and C_g will vary with t_1. All three will decrease with a reduction in t_1, but the above ratio will not vary enough to cause an appreciable error in the calculations.

Fig. 3 shows a series of curves illustrating graphically the foregoing calculations. Basic gas entering temperature was assumed as 2400 F and the effects of higher and lower entering temperatures are shown. A basic operating pressure of 400-lb gage was assumed. For 1000-lb pressure the effect on outlet temperature variation would be approximately 5 per cent greater and for 100-lb pressure approximately 5 per cent less.

Effect of Variable Operating Pressure

A further examination of Equation (7) will indicate that the pressure at which a boiler is operated affects not only the final gas temperature, t_2, but also the difference between final gas temperature and steam temperature, $(t_2 - t_s)$. This is obvious since t_s, which appears twice in the equation is dependent on the operating steam pressure.

A study of the effect of variation in operating pressure on $(t_2 - t_s)$ involves calculations similar to those previously discussed for the effect of entering gas temperature. Here again the magnitude of the effect on $(t_2 - t_s)$ of changes in t_s will depend on the magnitude of $(t_2 - t_s)$. This is illustrated in Fig. 4. For example, assume that a boiler when operating at 400-lb gage pressure has an outlet gas temperature 200 deg F above saturated steam and that, other things being equal, the pressure is increased to 1000 lb per sq in. Assuming a constant entering gas temperature of 2000 F, then the effect of this change will be $(2000 - 547)/(2000 - 448) \times 200 = 187.5$ deg F, a reduction of 12.5 deg F. However, had the boiler been of such design or operating at such a rate as to have an outlet temperature 400 deg F above saturated steam, then by a similar calculation $(t_2 - t_s)$ would become 375 deg F or a reduction of 25 deg.

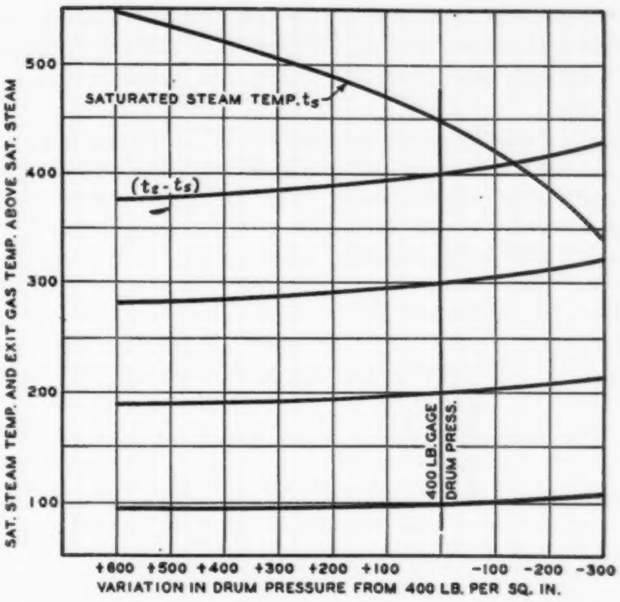

Fig. 4—Effect of variable drum pressure on saturated steam temperature and on exit gas temperature

The above calculations show why gas temperature test data or standard performance calculation of $(t_2 - t_s)$ based on a boiler operating at one pressure, cannot be used directly for any other pressure without considerable error, especially if the outlet temperatures above saturated steam are high. The assumption of 2000 F gas temperature entering the boiler for plotting the $(t_2 - t_s)$ curves in Fig. 5 introduces an error under some conditions. For example, if the gas temperature entering the boiler had been 2400 F, the effect of variation in operating pressure on variation of $(t_2 - t_s)$ would be approximately 20 per cent less; and would be approximately 35 per cent greater for 1600 F. Then referring to calculations in the previous paragraph, instead of a reduction of 25 deg F in $(t_2 - t_s)$ by increasing the pressure from 400 lb to 1000 lb the reduction would be 0.80×25 or 20 deg F with 2400 F entering temperature and 1.35×25, or 34 deg with 1600 F entering temperature.

Fig. 4 also shows a plot of saturated steam temperature, t_s, against operating pressure. It will be noted that as the pressure increases from 400 to 1000 lb per sq in., t_s increases from 448 to 547 deg F or 99 deg, but according to the other curves and the sample calculations given above, for the same increase in pressure the final gas temperature would increase only $99 - 12.5 = 86.5$ deg F if $(t_2 - t_s)$ were 200 deg F and only $99 - 25 = 74$ if $(t_2 - t_s)$ were 400 deg F with gas entering the boiler at 2000 F.

Effect of Superheat

Heat transfer to the tube surface of a superheater from the gases passing over it does not take place in accordance with Fig. 2 or Equation (7) because the steam within the tubes is not flowing at constant temperature. Superheaters may be designed for parallel-flow, counter-flow or, as is often the case, a combination of parallel and counter-flow of the fluids. Furthermore, superheaters may be designed to be placed in the furnace entirely external to the boiler proper or they may be placed within the boiler proper in such a way as to receive no luminous

radiant heat, or, as is often the case, they may be entirely within the boiler but at the same time partially exposed to radiant heat from the flame and refractory of the furnace.

The problem in calculating the effect of the presence of a superheater on the final gas temperature from the boiler, is first to determine the reduction in gas temperature caused by the superheater at the point within the boiler where the superheater is located. In a divided superheater which is part "radiant" and part "convection" the proportion of the total superheater absorption due to luminous radiation must be allowed for separately. This applies also to a superheater of the "inter-tube" type located within the boiler but partly exposed to luminous radiation. In the latter case the problem of allowing for the effect of the superheater is further complicated by the fact that the adjacent boiler surface is absorbing heat at the same time, and the mean temperature difference for the superheater is affected by convection (and non-luminous gas radiation) heat absorption by both boiler and superheater surface.

In the calculation of gas temperature drop across a superheater, we are dealing with boiler output on the one hand and input on the other. The heat absorbed by the superheater is not directly proportional to the rate of evaporation and degrees superheat obtained, but the weight of the gas stream supplying this heat is, for a given fuel, inversely proportional to the efficiency of the overall unit including burning equipment, economizer, air heater, cinder recovery, etc., and directly proportional to the excess air coefficient. For instance, a superheater must absorb 60 per cent more heat to give 100 deg F superheat at 1000-lb pressure than at 200 lb but only 30 per cent more to give 300 deg F superheat at 1000 lb than at 200 lb. Also, at 200 lb pressure a superheater must absorb 2.7 times as much heat for 300 deg F superheat as for 100 deg F but at 1000 lb it will give 300 deg F superheat with only 2.4 times the absorption required for 100 deg F superheat. For a given rate of evaporation and fuel, the weight of gases passing through the superheater may vary as much as 25 per cent due to variation in efficiency, and 50 per cent or more due to variation in excess air. These extreme variations will not, of course, be encountered on any one steam generating unit with any one fuel but rather they represent the possible extremes tending to introduce error if one is attempting to devise a simple method for calculating gas temperature drop across a superheater, or its resultant effect on boiler-outlet temperature, without making allowance for variations in fuel, burning equipment and heat recovery apparatus additional to boiler and superheater.

To illustrate the foregoing by a comparison of the use of two fuels, consider blast-furnace gas and coal. For a given rate of evaporation on a given steam generating unit the weight of gases from the former fuel will be approximately 75 per cent greater than from the latter, and the gas temperature drop across the superheater for a given heat absorption or degrees of superheat will be proportionately greater for coal than for blast-furnace gas. The ratio of flue-gas weight to steam weight is therefore an important factor in determining the effect of superheat on gas temperature drop across the superheater or gas temperature leaving the boiler.

Another factor to be considered is the location of the superheater within the boiler. This may be illustrated by reference to Fig. 6 or to Equation (7) in which t_1 may be taken as to the gas temperature entering the portion of the boiler following the superheater. A few simple calculations, similar to those given in previous paragraphs, will show that the effect on final gas temperature, of a given gas temperature drop caused by the superheater, is greater the nearer the superheater is located to the boiler outlet. In other words, the hotter the gas stream at the point where the superheater is located the less will be the effect of a given superheater heat absorption on the final gas temperature, other things being equal.

It is obvious from this discussion that the only accurate method of calculating the final gas temperature of a boiler having a superheater is to first use Equation (7) for the portion of the boiler preceding the superheater, then calculate the gas temperature entering the portion of the boiler following the superheater by subtracting the gas temperature drop in the superheater zone, and finally using Equation (7) for the remainder of the boiler surface.

Gas Temperature Gradient

It is often desirable to know the approximate gas temperature at various points within a boiler in order to select proper materials for baffles, baffle supports, superheater hangers, soot blower elements and hangers, etc. The usual method of determining these temperatures quickly is to plot gas temperature above saturated steam temperature as the ordinates on a logarithmic scale and percentage of surface passed over as abscissa on a uniform scale and connect $(t_1 - t_s)$ entering the boiler with $(t_2 - t_s)$ leaving the boiler (see Equation (7)) by means of a straight line to determine intermediate temperatures. The conditions under which this method may be used with reasonable accuracy are now discussed.

The following equation was developed in a preceding paragraph

$$(t_2 - ts) = \frac{(t_1 - t_s)}{\text{antilog}_e\left(\dfrac{S \times R}{K \times W_g \times C_g}\right)} \quad \ldots\ldots\ldots (7)$$

in which:
- S = surface of boiler or some particular portion thereof, sq ft
- R = average heat transfer rate for surface S, Btu/sq ft/hr/deg/F
- W_g = gas weight flowing over surface S, lb/hr
- C_g = mean specific heat of gases passing over surface S, Btu/lb/deg F
- K = radiation and leakage factor explained previously
- t_1 = temperature of gases entering surface S, deg F
- t_2 = temperature of gases leaving surface S, deg F
- t_s = temperature of saturated steam corresponding to operating pressure, deg F

This may be written:

$$\log_e(t_2 - t_s) = \log_e(t_1 - t_s) - \left(\frac{S \times R}{K \times W_g \times C_g}\right) \ldots . (8)$$

However, Equation (8) may be simplified by detailed consideration of the problem at hand. In the first place, we are dealing with a given boiler, operating at a given rating with a given furnace, fuel, method of firing, operating pressure, etc. Therefore, $(t_1 - t_s)$ and W_g are constants. Furthermore, if the boiler is of such design that it may be baffled for uniformly increasing mass velocity to give approximately constant linear velocity, then the ratio R/C_g will be nearly constant if

the surface is of the same general character and effectiveness.

Based on the above assumptions Equation (8) may be simplified to the following:

$$\log_{10}(t_2 - t_s) = -0.4343CS + \log_{10}(t_1 - t_s) \quad \ldots\ldots (9)$$

in which C = a constant combining R, W_g, C_g and K of equation (7) or (8)

S = the effective boiler-heating surface

This equation will be recognized as being of the familiar straight-line form, $y = mx + b$, having $\log_{10}(t_2 - t_s)$ for y (ordinate), S for x (abscissa) and the constant $\log_{10}(t_1 - t_s)$ for b (y intercept). The slope, m, of the straight line is negative and has a value of $0.4343C$ when $\log_e$ is converted to $\log_{10}$.

If a superheater is present within the boiler its surface may be included in S provided its heat transfer rate is approximately the same as that of the boiler surface preceding and following it. Many superheaters are designed, however, for a higher gas velocity and heat transfer rate in order to reduce the amount of surface required. In such cases the gradient may be determined as illustrated in Fig. 5.

This figure also illustrates how the gas temperature leaving a boiler with a superheater may be determined graphically from the established gradient for the same boiler without a superheater. The method is to reduce the gas temperature (on the vertical line shown in Fig. 5) by the calculated amount at a point on the gradient line corresponding to the location of the superheater. Then, if the boiler were so baffled as to satisfy the conditions required to make its gradient a straight line (see above), continuation of the gradient line after the superheater at the same slope ($0.4343C$) will involve very little error, or if the required accuracy warrants the time involved, the new slope after the superheater may be calculated.

Fig. 6 illustrates the statements made previously regarding the effect of superheater location on final gas temperature. For example, this figure shows the effect of a 300 deg F gas temperature drop across a superheater located behind about 40 per cent of the effective boiler surface, called Superheater "B." However, had the same superheat been obtained by means of a superheater located behind only 5 per cent of the boiler surface, called Superheater "A," the gas temperature drop across the superheater would have been approximately the same (assuming no luminous radiation) but the difference in effect on outlet gas temperature is shown by the

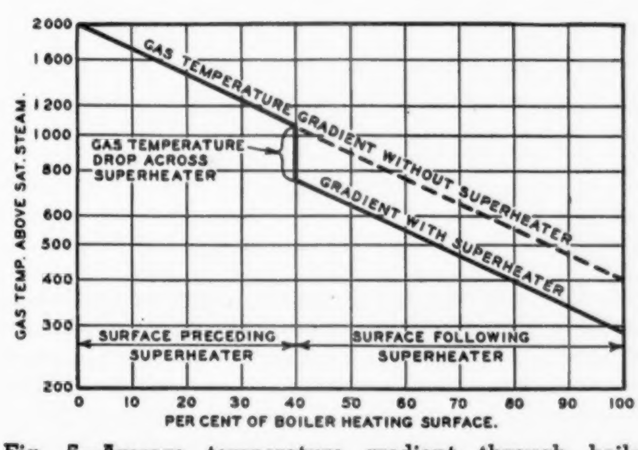

Fig. 5—Average temperature gradient through boiler showing graphic method of correcting for superheat

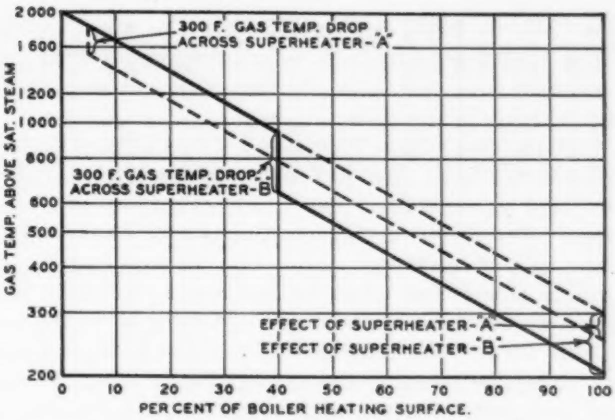

Fig. 6—Average temperature gradient through boiler showing effect of superheater location

figure to be much less. For the latter case the same gas temperature drop is layed off graphically at 5 per cent of the boiler surface and it is shown that, owing to the logarithmic ordinate scale, the reduction in final gas temperature caused by this superheater will be much less.

Analysis of Draft Loss

The various items which go to make up the overall draft differential between boiler inlet and outlet are "stack effect," loss of head due to turns, friction at right angles to tubes and friction due to flow parallel to tubes and baffles. All four are generally encountered in any one boiler in combination with constantly decreasing temperature and constantly varying (sometimes suddenly) velocity.

The stack effect in the furnace of a large semi-vertical boiler may be as much as 0.70 to 1.0 in. w.g. It is therefore important in the measurement and calculation of boiler draft differential to know or specify the location of the point of furnace draft measurement since the stack effect of the gases above this point may be subtracted from the calculated friction, and will have reduced the measured reading by the same amount. The point usually specified is at the top of the furnace midway of the length of the front boiler tubes. Stack effect in the several boiler passes is taken as plus for down-flow and minus for up-flow passes in the summation of boiler draft differential.

The turns encountered within boilers are of the worst sort in that they are generally 180 deg and very abrupt. There is no curved "throat" since the turns are generally made around the end of a baffle not more than 3 in. thick. In both cross-baffled and parallel-baffled boilers there is generally a change of shape at the turn although this may not mean a change in area or velocity. Therefore there is no fixed ratio of centerline radius to depth of passage. This ratio, however, will approximate 0.50 at which value the loss of head on a 90-deg rectangular bend may be taken as 100 per cent of the velocity head. This applies more accurately to ducts of nearly square cross-section. For ducts of high aspect ratio (i.e., ratio of width to depth) the turn loss for the same radius ratio will be reduced to 65 per cent of this for an aspect ratio of 3 and 50 per cent for an aspect ratio of 6.[5] For cross-baffled boilers the average aspect ratio will vary from approximately 2 for a 10-ft wide unit to 6 for a 30-ft wide unit. For parallel-baffled boilers it will vary

[5] See *General Electric Review*, June 1927, Loring Wirt.

from $3^1/_2$ to $10^1/_2$, respectively, for the same widths. These figures are, of course, only approximate since they vary with tube length in the first case and with depth of tube bank in the second.

The foregoing discussion suggests a method of approximating the turn loss by consideration of radius ratio and aspect ratio. The figures given are for 90-deg turns. For turns of 180 deg the loss of head will be twice that of a 90-deg turn. The obstructions in the form of tubes that occur in the bends are taken into consideration here only in the calculation of the net passage area. The velocity will not always be constant and should be taken as the average for entering, leaving and midpoint. Friction due to tubes must be allowed for separately.

Fluid friction parallel and at right angles to tubes is a subject that has received much study. For turbulent flow the loss varies nearly as the square of the velocity, approaching proportionality to the second power as a limit for higher values of the Reynolds number and rougher surfaces. Almost all gas-flow problems encountered in engineering lie in the turbulent region and it has been found that the Fanning equation in modified form can be used for flow either parallel or at right angles to tubes. A recommended form for fluids flowing outside of tube bundles is as follows:

$$P = Cf\frac{LM^2}{\rho D} \qquad (10)$$

in which P = pressure drop or draft loss, inches w.g.
C = a constant
M = mass velocity of gases, lb/hr per sq ft of gas passage area
D = equivalent diameter = 4 times free volume divided by the outside tube surface
L = length of path of gases
ρ = density of gases, lb per cu ft
f = friction factor which is proportional to D, M, Z, and will vary with the roughness (Z = absolute viscosity)

The use of "equivalent diameter" to take care of various tube diameters and tube spacing for cross flow should be limited, according to some investigators, to staggered tubes only and tube spacings not exceeding twice the diameter of the tube. The diameter, degree of stagger and geometric relation of tubes is very important when considering flow at right angles to tubes. As an illustration, if the tube spacing in the direction of flow across tube rows in alignment is greater than about four diameters, it is quite probable that the draft loss per row will be as great as across tubes of the same diameter and spacing when staggered. For spacings less than this the staggered arrangement will give the greater loss and it will bear some relation to spacing and degree of stagger. Tubes in alignment, or staggered tubes on wide spacing, must therefore be correlated in some other manner than the use of "equivalent diameter," or else use of data must be limited to the particular diameter and spacing for which it was determined.

Friction due to flow parallel to tube banks can be estimated from formulas for flow through pipe by substituting for pipe diameter the equivalent diameter of the boiler pass as defined in connection with Equation (10). Friction factors for flow inside pipes may be used, making due allowance for roughness in comparison with the kind of pipe for which the friction factors were obtained.

New Large Turbine-Generators on Order

Outstanding among units now under construction in the turbine shops of the General Electric Company at Schenectady are two 60,000-kw hydrogen-cooled turbine-generators for the Windsor Station of the Ohio Power Company and the West Penn Electric Company, at Beach Bottom, W. Va., and a 75,000-kw unit for the Cahokia Station of the Union Electric Light & Power Company, St. Louis, Mo.

The former will be the largest high-pressure, non-condensing, high-temperature machines yet built, and the generators will be the largest to operate at 3600 rpm, as well as the largest to be hydrogen-cooled. The machines are designed for 1250 lb gage pressure, 925 F total steam temperature, and 235 lb gage back pressure. They will be superimposed on the existing equipment.

The Windsor plant serves both the Ohio Power Company and West Penn Electric Company systems, and is jointly owned by the two companies. The new turbine-generators will be used to provide for increased demand and to improve the overall efficiency of the station. The units are expected to be in service late in 1938.

Hydrogen cooling as applied to turbine-generators is comparatively new. It has been used in the past, however, in many synchronous condensers. The 40,000-kw high-pressure turbine-generator ordered some months ago for the Logan Station of the Appalachian Power Company was the first to have hydrogen cooling.

Hydrogen cooling makes possible, economically, large generators at 3600 rpm which if air-cooled would have greater air friction and rotational losses. Hydrogen reduces these windage losses to 10 per cent of what they would be with air, and this results in an improvement in the efficiency of the generator of from 0.6 to 1.1 per cent, depending upon the size of the unit. Hydrogen requires considerably less pressure to circulate the necessary volume and it has many times the thermal conductivity of air. It also removes more heat from a given surface for the same temperature drop. Although hydrogen results in a slight increase in capacity for a given physical size, the electrical characteristics still determine the capacity of the unit. Because of the more expensive construction required with hydrogen cooling, its use cannot be justified in small sizes, but it makes possible large generators at 3600 rpm.

The capacity of the Cahokia Station of the Union Electric Light and Power Company of Illinois, at St. Louis, will be increased early in 1938 by the addition of the turbine-generator. This unit will be similar to another 75,000-kw turbine now operating in that station, which was installed in 1928. The unit, will operate at 315-lb pressure, 725 F and one inch absolute back pressure.

A. C. Fieldner, Chief Engineer, Experiment Stations Division of the U. S. Bureau of Mines was nominated for president of the American Society for Testing Materials at the recent annual meeting of that society. Doctor Fieldner has been associated with the Bureau of Mines since 1910 and has long been active in developing standard methods and apparatus for analyzing coal.

Unusual Boiler Problems Caused by Corrosion

By CYRUS Wm. RICE
Water Chemist and Engineer
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The author shows how oxygen corrosion in boilers, condensate return lines and accessory equipment can be prevented where the feedwater treatment in use is non-deposit forming and such as to eliminate protective coatings of calcium and magnesium over the water-exposed metal surfaces. The use of sodium sulphite as an inhibitor of oxygen corrosion is discussed, as is also the employment of catalytic agents.

ALTHOUGH problems of this kind are few and far between when compared to the many complex conditions which favor their development there are cases nevertheless that are allowed to continue because the remedies in use are inadequate or improperly applied.

Oxygen in the feedwater is the predominating cause for most of these puzzling problems, although some occur in which electric currents are found to be the underlying reasons back of their existence. In cases of the latter type the current source is traceable either to leakage from an electrical circuit, or to cell action between dissimilar metals in which the boiler concentrates act as the electrolyte.

In some rare cases the source of such currents has been known to start in nearby water-submerged dirt fills containing the essential constituents for setting up an electric cell. In either event the metal losses which result are caused principally by direct and not alternating currents. The extent and rapidity of this action is governed by the strength of the currents, being excessive between 200 and 500 millivolts and almost negligible under 5 millivolts.

Oxygen, as a cause for corrosion, proves most troublesome where the feedwaters are non-deposit forming, such as with the better conditioned zeolited waters. This is because of the constant tendency to dissolve any protective coatings of calcium and magnesium which may have formed over the water-exposed metal surfaces in boilers, thus keeping these surfaces open to the attacks of oxygen. The free and half-bound carbon dioxide originally contained in the raw water and which is carried along in the softening processes is most responsible for these properties.

Another influence governing this activity is the velocity with which the water within the boiler flows over the underwater metal-exposed surfaces and the influence this has in either preventing or allowing the oxygen to become attached thereto. Boilers, therefore, which are so designed as to allow for low velocities and are fed with water that does not provide a protective coating, as previously explained, offer the greatest obstacles to feedwater conditioning for the prevention of oxygen corrosion. In cases of this type, even deaerating heater operations with oxygen concentrations at zero by the Winkler test, and supplemental treatments of sodium sulphite in quantities in excess of practical requirements, have failed to prevent excessive corrosion in some instances.

With puzzling problems of this kind the final result becomes very serious if not corrected. However, they can be prevented—

First—where the information secured for control purposes is accurate and dependable under all conditions;

Second—where the treatments in use neutralize or destroy the dissolving action of the feedwaters;

Third—where the natural solids in the feedwater or those artificially included are permitted to maintain protective film coatings over the involved surfaces;

Fourth—where the sodium sulphite treatment is given more time for removing the oxygen from the feedwater before it reaches the boiler proper.

Some of the best information to be had for such control purposes makes it necessary to compare the relations between the chlorides in the boiler blowdown water (boiler concentrates) and sodium sulphite in this same water with the relation between these same constituents in the water supplied the boilers. As the direct titration of the feedwater in use for chloride determination is not dependable, especially when the chloride content of a feedwater is low, it is advisable, because of the required exactness of such relations, to resort to the use of more accurate determinations. The weighed or gravimetric practice best meets this qualification, because it eliminates all influences that have any tendency to mask the results. The next best method for accuracy is to boil down, say, 10 or 20 volumes of feedwater to one and then titrate for chlorides, dividing the product of this by the number of concentrations. With either method, the results are much to be preferred over the straight titration of a sample of feedwater in which the chlorides are low.

The value of the gravimetric method in this work is

probably best illustrated in Table I. Here we have a study which shows the influence the greater sulphite dosings have on the consumption of this chemical in the conditioning of the feedwaters for the removal of oxygen. The concluding facts in this are given in the last column and include the total pounds of sulphite destroyed per 1,000,000 lb of feedwater. By referring to this, it will be found that the loss varies with the sulphite dosing. We know part of this is caused in the sulphite feeding equipment, but with the better practices in use of sealing the chemical dosing solutions against air contamination, this loss holds fairly constant and relations as obtained by the method mentioned provide a very dependable guide for sulphite control purposes.

As a matter of further information to the reader, the sulphites in the blowdown water, such as those reported in Table I, are determined through a back-titration, using iodine and sodium thiosulphate. On the other hand, the sulphites for the feedwater are based on the actual weight of this chemical per 1,000,000 lb of water fed to the boiler. In either case, the results are comparable and final computations represent the nearest approach to accuracy that is available for this work.

To prevent the dissolving action of a feedwater, as previously mentioned, it is necessary to neutralize the contained carbonic acid. Caustic soda is generally used for this purpose, although caustic lime, when properly applied, proves the most reliable in obstinate cases.

The influence of the caustic soda treatment is largely controlled by the pH of the feedwater. On the other hand, the caustic lime additions in the amounts required for this work are often so small as to make little or no impression on the pH of the feedwater. The quantity actually required for the work is less than the total hardness that remains in the usual effluents from lime-soda hot or cold process softeners, and which is $^3/_4$ gr, or better, for most operations of this kind. The addition, therefore, of caustic lime to the product of a good zeolite operation should be less feared and much more flexible for coating water-exposed surfaces against the attack of corrosive gases than is the lime-soda softener product itself.

This application has special significance in the operation of high-pressure boilers at high ratings where lime carryovers into the steam-producing sections may cause troublesome deposits. Again, the location for introducing the lime in a feedwater system is very important if the surfaces affected are to be protected uniformly. As the colder waters possess the greatest solvent properties for caustic lime, the most logical procedure for getting the better distribution is to introduce it into the cold water before the heater. Any cold surge or storage tank can be used for this purpose. The situation with respect to the use of lime is further improved where the time for dissolving this chemical can be increased before it reaches the hot-water heating and boiler-feeding equipments.

In either event, the practice which insures the greatest protection through artificially produced depositions is that which permits the lime to dissolve to the greatest extent possible and then react with the free and half-bound CO_2 in the feedwater before it reaches the economizer sections where corrosion is most active. In this same connection and for the purposes discussed it is, also, very important that the lime carry the maximum calcium solubility.

The protecting influence of both caustic treatments and the work of the sulphite may be further checked by the periodic location of sensitized steel rods and plates within the feedwater loop and affected boiler sections and the later removal of these for analysis and inspection. The accompanying exhibits of detector plates gives a very good account of these influences in preventing corrosion.

Referring now to the third heading of this subject, practical experience in the field has definitely shown the necessity in a few scattered cases of carrying pH values in the feedwater near 10.5 before the dissolving action of some waters can be fully corrected. This is in line with practice now in use in which equivalent or better pH values are maintained in the boiler proper for the same purpose.

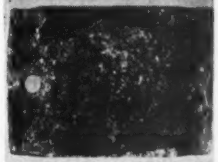
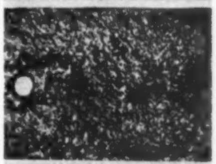
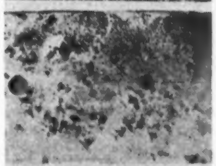
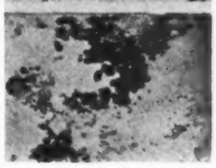
In confirming a previous statement, it may be emphasized here that lime, when rightly controlled, proves the best influence in holding the losses resulting from corrosion to the possible minimum. This is due to its double properties of fixing the CO_2 as a solid deposit and of placing the product of this reaction as a protective film of calcareous material over the metal-exposed surfaces.

Where the use of caustic lime does tend to cause unsatisfactory sludge depositions in the steaming sections of boiler, the effect can be largely prevented through supplemental colloid treatments, and with better results where the soda alkalinities within the boiler are kept under the concentration limits which retard their activities. This is important because colloids actually lose their properties for removing sludge from the boilers in cases where the boiler soda alkalinities are excessive. On the other hand, colloids improve their properties as the alkalinities decrease, being greatest in the lower brackets.

Another very necessary influence in either curbing or

TABLE I

Period	Blowdown Water NaCl	Blowdown Water Na_2SO_3	Blowdown Water Na_2SO_3/NaCl	Feedwater NaCl	Feedwater Na_2SO_3	Feedwater Na_2SO_3/NaCl	SO_3 Loss 1 ppm NaCl	SO_3-Total Loss per 1,000,000 lb of feedwater
1/23/36 to 2/23/36, inclusive with pH 10.5. Increased through NaOH additions to pump suction boiler feed. Recirculation blowdown discontinued	178	112	0.63	3.28	3.60	1.10	0.47	1.53 lb
2/24/36 to 3/10/36, inclusive. Includes previous NaOH and CaOH plus sea moss and gum	155	90	0.58	3.00	3.54	1.18	0.60	1.80 lb
3/11/36 to 4/9/36, inclusive. Includes previous NaOH, CaOH, Sea moss, gum plus 50% SO_3, inclusive	134	165	1.23	2.16	5.3	2.5	1.27	2.70 lb
4/10/36 to 4/30/36, inclusive. Includes previous treatments plus PO_4 increase and recirculations	170	194	1.14	2.21	5.17	2.34	1.20	2.65 lb
5/1/36 to 5/10/36—SO_3 unchanged. All other treatments the same	182	184	1.00	2.18	5.4	2.5	1.5	3.20 lb
5/11/36 to 5/17/36, inclusive. SO_3 reduced from 5 to 3 rate; lime discontinued	155	131	0.84	2.12	3.15	1.5	0.66	1.40 lb
5/17/36 to 5/28/36—SO_3 reduced from 3 to 2 rate. Lime renewed May 24.	149	96	0.65	2.08	2.25	1.08	0.43	0.90 lb

No.	Steam Drum Dectector	Days in Service	Weight Loss, gr/100 days	Treatment	Remarks
1		100	0.0170	internal	plate is in excellent condition showing no calcareous deposit nor any form of pitting or corrosive action.
2		83	3.0700	internal	plate shows a very corrosive action, manifesting itself as wasting.
3		147	0.4900	zeolite softener	plate shows positive pitting action caused mainly by CO_2.
4		359	0.4440	zeolite softener	plate shows corrosive action caused principally by CO_2.
5		76	0.2210	hot lime soda	plate shows a few pit marks, indicating some corrosive action, which however is very light. A light calcareous deposit is noted.
6		136	0.0215	cold lime soda	plate shows a light lime deposit, but no corrosion.

Detector plates showing influence of caustic treatments and sulphite

correcting difficult corrosion problems lies in the use of sodium sulphite. This can be made more effective where its contact with the feedwater is increased after the water is deaerated, or where catalysts are employed in accelerating the reactions between the sulphite and oxygen in the feedwater.

According to operating facts, sodium sulphite proves to be a very sluggish agent in its removal of the last traces of oxygen, even in large excess doses and under the most favorable temperature conditions. Any means, therefore, that will increase both the time and contact of this chemical with the oxygen in the water increases its effectiveness. This can be accomplished to some extent by introducing the sulphite into the suction chamber of the heater, preferably recirculating it with the water in this section, using an outside manifold and several jet heaters for this purpose.

Further improvements in this reaction can be brought about through the use of catalysts. Laboratory experiments have clearly indicated the value of the latter in increasing the rapidity with which oxidizable sulphur compounds go to sulphates through their absorption of oxygen. Some of these materials, under laboratory setups, have proven to be eighteen times more effective than others. Table II indicates the comparative values of a number which were tried out in this way:

TABLE II

	Relative Values	Remarks
Catalyst A	+17.6	Nitrogen compounds
B	+16.3	Nitrogen compounds
C	+14.0	Nitrogen compounds
D	+11.2	Nitrogen compounds
E	+14.3	Nitrogen compounds
F	+14.0	Rare metal chlorides
G	+12.6	Nitrogen compounds
H	+ 9.6	Manganese sulphate
I	+ 9.6	Nitrogen compound
J	+ 8.0	Iron sulphate
K	+ 9.5	Copper sulphate
L	+ 9.4	Nitrogen compound
M	+ 7.5	Vanadium disulphate
N	+ 5.6	Aluminum sulphate
O	+ 4.2	Lead nitrate
P	+ 3.9	Zinc sulphate
Q	+ 2.0	Barium chloride
R	+ 2.0	Potassium chromate
S	+ 2.4	Tin chloride
T	+ 2.4	Uranium nitrate
U	+ 0.6	Calcium chloride
V	+ 0.8	Strontium chloride
W	0	No catalyst
X	− 1.2	Sodium tungstate
Y	− 2.1	Sodium arsenate
Z	− 1.5	Sodium molybdate

A, B, C, D, E and G are nitrogen compounds of the inorganic salts in which two salts were combined in one experiment. While this was characteristic of most of the nitrogen compounds there were exceptions to this as experiments O and T indicate. Here the catalytic effect of the chemicals in use was very low. On the other hand, the X, Y and Z experiments with soda salts were negative in their effect.

Up to the present time the use of catalysts in this spe-

cial application has been carried out in the field in a very limited manner, having been confined almost altogether to the use of copper sulphate. The writer has never definitely measured the value of this one addition as its use in this particular connection was the result of other recommendations. It was finally discontinued because of a specific case in which severe electrolytic corrosion developed as the result of some under-water depositions of metallic copper.

The application of the other materials in the field and which have proven more effective in the laboratory set-up is now under way and should develop some very interesting facts for a future report. In the meantime, the judicious use of lime, maintenance of the desired pH in the feedwater and increase in both time and extent of the agitation of the sulphite with the feedwater before it enters the boiler provide the best measures for correcting nasty boiler corrosion problems caused by oxygen.

What has been said about the conditioning of feedwaters for correcting difficult corrosion problems within the boilers themselves also applies, with few exceptions, to similar situations within condensate return lines and steam heating systems. The principal of these exceptions is the influence the treatments in use for preventing boiler scale depositions have on the corrosive properties of the steam and equipments it supplies. This can be excessive and is often the case where the conditioning chemicals themselves are the source of additional corrosive gases in the steam leaving the boilers. This is an important consideration in control work and should not be lost sight of in the treatment of boiler feedwaters for preventing scale and corrosion.

In any event, the final solution of all boiler water problems can only reach a satisfactory conclusion where the facts for control work are dependable and knowledge covering their use carries the judgment of the better laboratory and field experiences in this special work.

New Research Plan at Battelle

The Trustees of Battelle Memorial Institute have announced a new plan to extend the scope and utility of the educational features of the Institute's basic program of scientific research. Since its inception, the policy of Battelle Memorial Institute has been to combine a program of fundamental research of general interest to industry with the applied research which it has carried on for individual companies and associations under its plan of sponsored research.

A new division of Research Associates has been established to supplement the work in fundamental science of the regular technical staff. The purpose of this new division is to offer intensive training in practical research to the best of the younger workers in selected branches of chemistry, metallurgy, fuels and ceramics. Appointments as Research Associate will be made for one year's duration and may be extended for a second year. A Research Associate will be expected to devote his entire time to a research problem approved by the Director and supervised by members of the Institute staff. The results of these researches will be published in order to contribute information that will be useful to science and industry.

Revised Standards for Coal and Coke

At the recent A.S.T.M. Annual Meeting, three detailed committee reports were presented on the subject of coal and coke and gaseous fuels. The Committee on Coal and Coke announced that it is planning to prepare a method of test for dustiness of coal and coke, since it is desirable to have a standard test method to determine the efficiency of treating coal and coke with chemical and oil compounds to allay dust and give a clean fuel.

Results obtained from check determinations made by a number of different laboratories of coal-ash fusibility determined by the Barrett coal-ash fusion furnace and by the gas furnace now specified in the Standard Methods of Laboratory Sampling and Analysis of Coal and Coke were announced as sufficiently promising to justify further studies so that the new furnace can be used as permissible equipment in the standard test.

A modified procedure to determine volatile matter of sub-bituminous coal, lignite and peat is contained in the Standard Methods of Analysis, in which these fuels are given a preliminary heat treatment to prevent rapid evaporation of moisture and gases. In order to bring out the fact that certain low-temperature cokes, green cokes, chars, anthracites and semi-anthracites should also be given such preliminary treatment and to include more detailed instructions for this modified method of determining volatile matter, a revision of the Standard Methods was approved by the A.S.T.M. meeting for publication as tentative.

The Committee also agreed to recommend to the Society an important revision of the present Standard Method of Sampling Coal. This revision gives a detailed procedure for the rapid reduction of gross samples of coal to a convenient quantity for transmittal to the laboratory. This is accomplished by mechanical methods of crushing and by riffle samplers. It is believed that this mechanical method of reducing coal samples will be widely used by coal producers and consumers who have occasion to do much coal sampling, as it is much more rapid and less expensive than the hand method described as the present Standard Method. The mechanical method supplements but does not replace the present hand method which is suitable for use by those who only occasionally have to sample coal.

A proposed method developed by the Sectional Committee on Classification of Coals, which is sponsored by A.S.T.M. under A.S.A. procedure, covering the designation of coal sizes when the results of screen analysis tests are taken to represent the condition of the coal as sold, was approved as a tentative method. This does not cover the standardization of screens used in the commercial preparation of coal but applies only to natural continuous ranges of sizes as produced by mining, handling, crushing, screening, etc. The Sectional Committee announced that while sufficient information had not yet been obtained to justify the formulation of specifications for the classification of coals according to type, it had reached tentative agreement that the types of coal should be classified into two groups, namely, banded and non-banded coals, and that these two groups should each be subdivided into three classes making six classes of coal types. It is hoped that in another year tentative specifications for type may be submitted for consideration.

STEAM ENGINEERING ABROAD

As reported in the foreign technical press

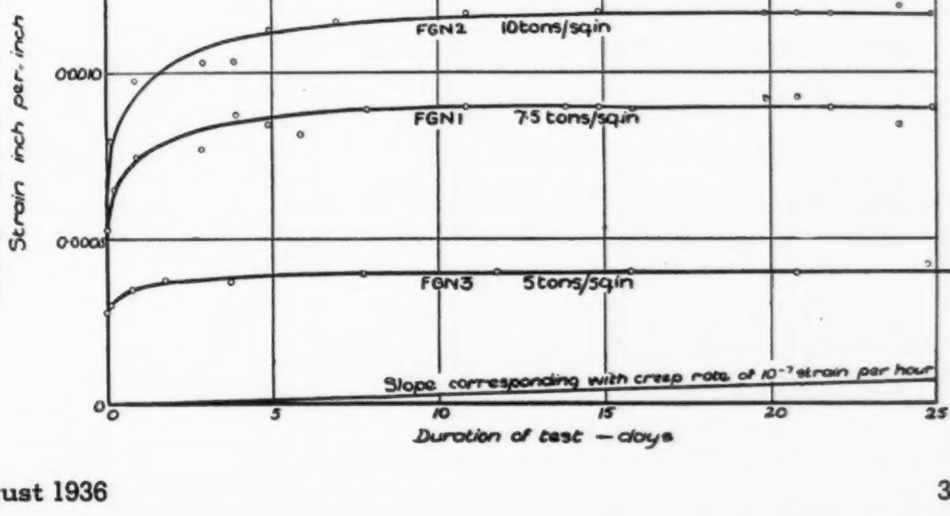
Stud-bolt Material for High Temperatures

In *The Steam Engineer* (London), for July, Douglas Wilson, Development Engineer of the United Steel Companies, Ltd., discusses the influence of high temperatures on stud-bolt steel and gives curves for long-time creep tests on "Durehete" (a chrome-molybdenum steel) at 896 F. His comments are in part as follows:

"Dealing with the development of steels for use as stud bolts, one of the first attempts was in the application of nickel and nickel-chromium steels. It was soon found, however, that while such steels had reasonably good creep-resisting properties, the effect of stressing at temperature had a marked influence on their toughness and they became exceedingly brittle in a relatively short period. Additions of molybdenum were then made to such steels in an attempt to inhibit embrittlement, but it has been shown that this element only serves to delay the time at which these steels begin to lose their notch-toughness.

"Research has proved conclusively that under conditions of stress at high temperatures, nickel-containing (martensitic) steels all suffer from this defect, and a steel has been developed, not only capable of resisting creep under practical conditions of temperature and stress, but also free from the defect of strain-age embrittlement. The material in question, known as "Durehete," is a chrome-molybdenum steel which was developed in the Central Research Laboratories of the United Steel Companies, Ltd."

The accompanying strain-time curves plotted from long-time creep tests on this material show that there are two distinct phases of creep. In the initial phase there is a rapid rate of creep within the first few days and this is followed by a slower rate which becomes practically uniform. The slope of the curve during the secondary phase for any period is expressed in "inches per inch per hour." This creep rate can be assumed to remain practically constant throughout the useful life of the material, provided the constitution of the material remains unchanged under the prolonged influence of temperature and stress.



Strain-time curves from creep tests on Durehete

Power from Refuse Destructor

A refuse disposal plant which generates a considerable amount of by-product steam for power was recently placed in service at Huddersfield, Eng. As described in the July 16 issue of *Electrical Times*, the incinerator process is divided into three stages, namely, (1) the pre-drying stage, carried out on an inclined moving stepped drying hearth; (2) the ignition, effected on an inclined stepped moving link-type burning grate; and (3) the stage of final combustion for which an inclined refractory-lined rotating cylinder is employed. The waste-heat boiler is of the four-drum type generating steam at 200-lb pressure and 600 F. The installation is designed for an output of 132 million pounds of steam per hour which goes to the Huddersfield Corporation Electricity Department located nearby.

The refuse has a moisture content of $24^1/_2$ per cent, a calorific value of 4664 Btu per lb as charged and on test an equivalent evaporation from and at 212 F per pound of refuse burned was 3.19 lb. It is estimated that 391 kwhr will be produced per ton of refuse and that the plant will turn out about eleven million kilowatt-hours per year.

6000-Kw Electric Boiler

Mention was made in these columns about a year ago of a new design of electric boiler brought out by Brown Boveri Company in which the water is delivered into the heating chamber through a series of centrally located nozzles, the water jets from which impinge on the electrodes and conduct the current. A boiler of this type,

rated at 6000 kw, 16,000 volts, has lately been installed in the plant of the Lonza Electric Supply and Chemical Works Company at Visp, Switzerland, and is described in the July 17 issue of *Engineering*. It is capable of producing 17,600 lb of steam per hour at 170 lb per sq in. pressure when supplied with raw water having a hardness of 10.4 deg and a specific resistance of 500 to 600 ohms per cubic centimeter.

Performance at St. Denis

In reporting the recent Electric Power Producers Conference at Scheveningen, *Engineering* of July 24 quotes from a paper by Robert Boudraut giving operating results of $2^1/_2$ years for the extension to the St. Denis Power Station in France. This extension consists of six pulverized-coal-fired B & W boilers supplying steam at 57 kg per sq cm (810 lb per sq in.) and 470 C (878 F) to three 50,000-kw turbine-generators. Atritor type mills are employed. The upkeep on the mills has averaged 1 franc (approximately 6 cents) per ton of coal pulverized and with dry friable coal the power required per ton of coal ground is a little over 20 kwhr. With coal that is hard and wet the consumption is about 17 kwhr per ton. With coal containing ash which fused below 1200 C (2192 F) the Bailey walls at times became covered with a coating of fused ash but this difficulty was overcome by using coal having an ash-fusing temperature over 1350 C (2462 F) and by the provision of auxiliary air inlets in the lower corners of the combustion chamber.

Despite a low load factor, the average thermal efficiency of the station has been 24.8 per cent while for periods of nearly constant load of 85,000 to 90,000 kw the efficiency has been 27.5 per cent.

Treatment of Flue Gases

Writing in the July 23 issue of *The Electrical Times* of London, Arthur Briggs and D. Northall-Laurie review existing methods of removing acidity in stack gases and discuss a new method which is now under experiment.

The one commercial process now in actual operation in two or three power stations, which has received considerable publicity, depends upon the use of a vast quantity of water. Another method about to be tried out on a commercial scale depends upon the use of water in a closed circuit together with the use of hydrated lime in suspension. In this the calcium sulphate is removed from the circuit in the form of sludge and the water is circulated again after the addition of more lime. A disadvantage of methods involving gas scrubbing is that the gases as finally discharged are saturated with water vapor at a comparatively low temperature and under certain atmospheric conditions may settle to the ground with little dilution.

The authors have been engaged for some time in experimental work on a dry process at a power station of the Yorkshire Electric Power Company. At first, the addition of lime to the coal was tried but it was found that from three to five times the theoretical amount of alkali was required which indicated that only a portion of the lime acted directly on the gas. Tests using finely pulverized magnesium limestone gave similar results.

Next the alkali was injected into the combustion chamber in the form of a dust cloud but here again too much lime was required, although there was some improvement in the drop in acidity of the stack gases.

It is now proposed to try out a combination of the wet and dry method. This consists of injecting into the flues through atomizing nozzles, at a location sufficiently hot to be above the boiling point of water, a slurry of slaked lime and water, only sufficient water being used as can be held in the flue gas without condensation at the prevailing stack temperature. These experiments will shortly be undertaken at the above-named station.

Water Softening on the "Queen Mary"

The water softening plant for treating boiler makeup on the "Queen Mary," as described in the July issue of *The Fuel Economist* (London), consists of a combined lime and base exchange system operating on the dual principle, with closed pressure sand filters. The plant has a normal duty of 300 tons (672,000 lb) of water per 24 hr. Two reaction tanks and a mixing tank are provided into which the raw water is pumped, while there is added the required amount of lime solution. The latter is prepared in a reagent tank, fitted with a mechanical agitator, and holds an eight-hour supply at the maximium rate of 2800 gph.

In the reaction tanks a maximum of $2^1/_2$ hr settling is given, and the sludge, which separates by gravity, is discharged to the bilge by means of a rotary collector gear at the bottom of each tank. The water then passes through the closed pressure quartz sand filters. Cleaning of the sand is effected by means of compressed air using a steam injector and this operation requires only five minutes.

The clear water then passes to two base-exchange softeners. The amount of salt required to revive the base exchange material averages 4 lb per 10,000 grains of hardness removed.

Electricity Supply Progress in France

One does not read much these days concerning power developments in France probably because little in the line of spectacular or outstanding installations have been made there recently. However, it is apparent from statistics published in the July issue of *World Power* that the use of electricity has gone forward in the past 12 yrs, although to a lesser extent than in England or the United States. The total output of electric generating stations in 1935 was 15,400,000,000 kwhr which was only slightly above that of 1930, the peak year. The average consumption per customer increased from 163 kwhr in 1923 to 319 kwhr in 1935.

The present generating capacity is $10^1/_2$ million kilowatts of which 66 per cent is steam and 34 per cent hydro. Capacity has increased faster than consumption and during the past few years the ratio of hydro to steam power has increased. The reason for installed capacity having increased faster than output is explained by the extremely variable load with high peaks; for instance that of Paris, which is typical, varies from 50,000 kw at 3 a.m. to 340,000 kw at 5 p.m.

REVIEW OF NEW BOOKS

Any of the books reviewed on these pages may be secured from Combustion Publishing Company, Inc., 200 Madison Ave., New York

Boiler Feed Water Treatment

By F. J. Matthews

This book not only covers the well-established processes of water softening and the usual methods of control, but it also gives an easily understood account of the newer methods of treating boiler feed waters which have been developed during the past ten years. These latter are designed to prevent scale formation, in particular in modern high-pressure boilers, where a water softened by the ordinary lime-soda process is in use.

The introductory chapters follow conventional lines. The characteristics of various natural boiler waters are described. The chemical principles of lime-soda water softening are then discussed, and a short account given of the constructional features and methods of operation of lime-soda plants. The beneficial action of sodium aluminate as a softening agent is explained in that a reduction is effected of the "residual hardness" of a lime-soda softened water, the formation of flocculent precipitates facilitates rapid sedimentation and filtration and the formation of silicate scale is retarded if not entirely prevented. Short sections are devoted to zeolite softeners, the treatment of alkaline waters, the use of condensate as feedwater, and water blending and lime-barium softening. The mechanism of scale formation and the meaning and purpose of carbonate and phosphate conditioning are well and simply presented.

The foregoing matters are dealt with in the first 96 pages of the book. The remainder of the volume is divided into separate sections devoted, in turn, to corrosion, foaming and priming, and analysis and routine testing.

An appendix comprises directions for the preparation of the standard solutions required in analysis, Clark's table of hardness, and appropriate chemical factors.

Boiler Feed Water Treatment contains 256 pages, size $5^1/_2 \times 8^3/_4$, including a comprehensive index. Price $5.00.

Cause and Prevention of Steam Turbine Blade Deposits

By Frederick G. Straub

This 52-page bulletin recently issued by the Engineering Experiment Station of the University of Illinois contains the report of an investigation made by Professor Straub in cooperation with the Utilities Research Commission Inc. to discover the underlying cause and corrective methods for turbine blade fouling due to solids carried over with the steam. The conclusions from the investigation, which is fully described in the bulletin, were that the basic material causing fouling is sodium hydroxide; that certain inorganic salts, such as sulphates, chlorides and carbonates, if present in sufficient amounts, will prevent this fouling; and finally, that certain organic salts, such as pyrogallol, sodium benzoate, and sodium gallate, if present in sufficient amounts will also act as a preventative.

Price of the bulletin 55 cents.

Symposium on Industrial Fuels

The *Symposium on Industrial Fuels*, recently published by the American Society for Testing Materials, comprises four extensive technical papers, as follows:

Coal and Coke; Occurrence, Testing and Utilization by A. C. Fieldner and W. A. Selvig

Industrial Fuel Oils by H. V. Hume

Manufactured Gas by P. T. Dashiell

Liquefied Petroleum Gas by W. H. Bateman

The authors of this symposium have followed a prescribed outline so that the papers have considerable comparative value. Among the subjects covered are the following: Historical background, magnitude of the industry, future availability of raw materials, possibility of new processes, tests which are applied to the materials, significance of tests, utilization of materials, general economic aspects of the product and utilization.

This pamphlet contains 70 pages, size 6×9. Price 75 cents.

Oil Heating Handbook

By Han A. Kunitz

Mr. Kunitz, a well-known authority on oil heating in the domestic and industrial fields, has translated the technical data of this growing industry into easy, everyday language. This comprehensive manual will be of interest to every man who designs, installs, sells or uses oil heating equipment. It answers the many questions home owners ask about oil heating, explains thoroughly and impartially the principles, installation and maintenance of oil burners, and provides students, oil burner men, as well as home owners, with simple, concise definitions of words used in connection with heating engineering.

Starting off with a short introductory chapter and a description of a typical oil burner installation, the book is divided into four main divisions: *Fuels* and *Combustion; Heating* (including Operative Controls); *Installation* and *Operation*; *Retail Selling*.

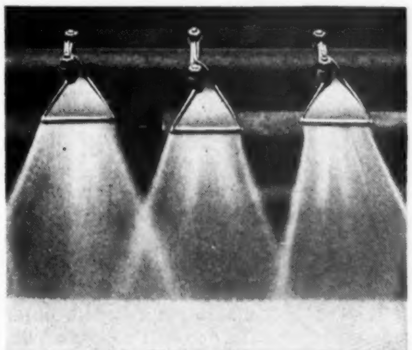
The book is profusely illustrated and contains many valuable charts and tables.

The *Handbook* contains 456 pages, size $5^1/_2 \times 7^3/_4$, including a comprehensive index. Price $3.50.

NEW EQUIPMENT

New Spray Nozzle

Link-Belt Company, Chicago, has developed a simple, non-clogging spray nozzle for a wide range of applications, including the cleaning of traveling water-intake screens. It is described as a scientifically shaped, smoothly polished curved bronze deflector with U-bolt for clamping the deflector securely to the water pipe, in such a position that it is just above the orifice (a plain drilled hole) in



the pipe wall. The width of deflector permits a comparatively large water jet, and thus allows fairly large dirt particles to pass through the orifice without clogging it. To assemble the deflector on the pipe, it is only necessary to place it in proper position over the orifice, and then tighten the hex nuts on the U-bolt. It is made in orifice diameters from $^5/_{32}$ to $^1/_2$ in. for pressures ranging from 20 to 100 lb per sq in. and in capacities up to 46.4 gpm.

Nalco Densimeter

The Nalco Densimeter or small floating bulb, recently put on the market by National Aluminate Corporation, Chicago, is a convenient and accurate method for determining boiler water concentration by the density method. Because it is not necessary to have a glass stem and be weighted by lead or mercury as is an ordinary hydrometer, the densimeter is exceedingly light and sensitive to changes of concentration of solids in the boiler water. This increase of sensitiveness gives more

accurate determinations. Also because of its size, it is conveniently handled and carried.

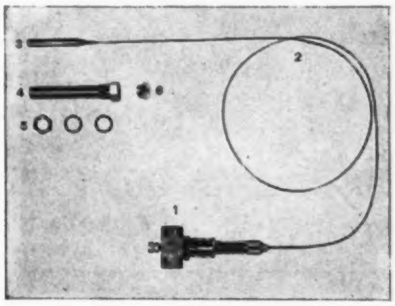
The small floating glass bulb is placed in a beaker of warm boiler water where it will sink to the bottom of the sample. The water is stirred with a special thermometer graduated in grains per gallon, and as the water cools a point is finally reached when the bulb will float. The concentration of the boiler water is then read direct from the graduated thermometer. The range is 25 to 500 gr per gal.

The glass bulb is encased in a metal container that can be attached to a watch chain and the thermometer has a clip so that it may be carried in a pocket.

Automatic Steam Control Valve

The Yarway Automatic Steam Control Valve shown is being manufactured by Yarnall-Waring Company, Philadelphia, and is designed for connection to the outlet or condensate end of steam heating coils in water heaters and in various types of industrial process equipment, in which close regulation of temperature is required. When so connected, this valve eliminates steam regulators and steam traps and frequently does away with pressure-reducing valves. It may also be used on the inlet end of the coil.

Valve (1) is of throttling type, operated by thermostatic element (2) of bulb and



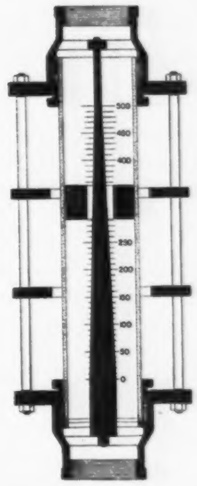
bellows design. The temperature-adjusting sleeve covering the bellows end of the thermostat is screwed in for lower temperatures, out for higher temperatures. The choke screw on the end of the valve is adjusted to suit normal condensate flow. The plug on the side permits check on operation. After initial adjustments for temperature and flow are made, the control is completely automatic in action, opening to discharge condensate when the temperature of the liquid falls a few degrees below that desired and closing when sufficient steam has been admitted to the heating coil to re-establish operating temperature.

Thermostat bulb (3) may be clamped to the outside of the heater shell but is usually inserted in sleeve (4) which projects into the water space in the tank. The sleeve is screwed into $^1/_2$ in. connection on the tank wall or fastened in place by a locknut and lead washers (5). The bulb is locked into the sleeve by split nut (6).

Taper Pole Rotameter

When oils and similar semi-opaque liquids must be metered, the standard tapered tube Rotameter is sometimes not completely satisfactory due to the difficulty of seeing the rotor through the layer of liquid between the tube and the rotor.

For such service Schutte and Koerting Co., Philadelphia, has developed a special



rotameter as illustrated. In place of the tapered glass metering tube a straight cylindrical tube is used and inside it is a tapered pole. The pole has its larger end at the bottom and the rotor fills the space between the tube and the larger end of the pole with only slight clearance. Flow of the semi-opaque fluid pushes the rotor upward in the normal manner but creates a free space between rotor and pole rather than between rotor and tube.

Since the film of liquid between the rotor and the tube is quite thin, the rotor can be seen without difficulty.

Stress Reliever for Welded Pipe Joints

When a welded joint is subjected to fatigue or normal service strains, residual welding stresses may become serious especially when they coincide with stresses resulting from internal forces. Hence residual welding stresses should be relieved to as great a degree as possible before the joint is put in service. The A.S.A. tentative Standards for Pressure Piping sets $^3/_4$ in. as the lower limit of wall thickness for which stress relief is essential.

The Detroit Electric Furnace Company has recently put out a stress reliever consisting of a portable low-frequency heating coil and protective insulating shield, both hinged so that they can be readily clamped around the pipe weld. By this means heat is developed in the weld metal and in the necessary length of pipe. A tap-change transformer with auxiliary

equipment for controlling the electric energy to the heating coil and controlling pyrometer is mounted on a portable hand truck. This equipment in addition to the tap transformer also includes a magnetic contactor which energizes or de-energizes the coil to give the desired temperature

control, a line safety switch, a control switch, a meter for reading the heating-coil current and a potentiometer-type indicating controlling pyrometer. In operation the primary cable is connected to any 60-cycle supply of approximately 100 kva capacity; the thermocouple is attached to the weld and to the controller; the coil is mounted around the weld to be relieved; the secondary cables are connected to the coil; and the correct voltages are secured by making the proper connections to the tap changer. The controller is then set at the desired temperature and the line and control switches closed. In order to limit the rate of temperature rise in the weld metal the starting voltage is ordinarily less than that required to bring the weld to maximum temperature and each subsequent advance is made by changing the tap connections at the transformer. When the temperature has been raised to its final value it is automatically maintained within close limits by the pyrometer- and thermocouple-controlled magnetic contactor

Automatic Pressure Compensator

In measuring the flows of compressible fluids such as steam, air and gas, variations in the static pressure of the flow directly influence the flow reading. To prevent errors from this source, The Foxboro Company, Foxboro, Mass., has developed an automatic pressure compensator for use with its flow meters.

The automatic pressure compensator mechanism, as shown in the illustration, is a simple spiral pressure tube connected to a lever. As the line pressure changes, the spiral adjusts the multiplication of the linkage between the float and pen arm. This is not an additive correction but a percentage correction; the linear amount of correction differs at different points of the chart and is calibrated to fit the flow formula of the meter. Thus a continuous, instantaneous and accurate compensation for pressure changes is obtained automatically.

At zero flow, pressure changes do not affect the position of the flow pen; at all other points on the chart the shift in pen position is just sufficient to compensate for the effect of this factor in the flow formula. On square-root flow scale meters the pen position is varied proportionately to the

change in absolute pressure; on uniform flow scale meters, pen position is varied proportionately to the square root of change in absolute pressure. Should maximum flow and maximum pressure occur simultaneously, the pen will not travel beyond the upper chart limit.

The Benson Boiler

In the contribution by Dr. F. Michel in your July issue, describing the Benson super-pressure steam generator, it is stated that "the Benson boiler and other special boilers were developed in Germany and to some extent in Switzerland."

Would you allow me to point out that this is totally incorrect. Mark Benson, with whom I was in personal contact for a considerable period in London, told me that he thought out the whole principle of the boiler while in the United States because of his long experience with the use of high-pressure steam in oil cracking, and the first Benson generator in the world was erected and operated experimentally at the works of the English Electric Co. Ltd. at Rugby (Warwickshire), a well-known town about 80 miles north of London.

Also I may say that I read the first paper on the Benson generator, in London before the Institution of Chemical Engineers, although descriptions had previously appeared in the technical press in the United States.

It will be remembered also that one of the greatest pioneers in the world of super-pressure steam generation was Jacob Perkins, who many years ago carried out detailed experimental work both in the United States and in England, and is believed to have reached pressures of over 1500 lb per sq in.

David Brownlie

London, Eng.

Sales of Boilers, Stokers and Pulverizers Show Marked Increase

According to statistics compiled by the Department of Commerce, the total number of power boilers sold during the first six months of this year was 1643 compared with 941 for the same period of 1935 and 829 in 1934. Mechanical stokers installed under power boilers during this six-month period totalled 996 for 1936, 778 for 1935 and 738 for 1934. The totals for pulverizers were, respectively, 207, 84 and 49. These figures may be regarded as an index of the present activity in the steam plant field.

Business Items

The Edward Valve & Mfg. Co. Inc. has appointed Tomlinson Steam Specialty Co., 1603 St. Clair Ave., Cleveland, as its sales representative in that territory.

The C. O. Bartlett & Snow Company of Cleveland announces the appointment of P. K. Reed to its engineering and sales staffs. Mr. Reed has long been associated with the field of coal and ash handling and was formerly chief engineer of the R. H. Beaumont Company.

Bernitz Furnace Appliance Company of Boston has appointed Henry F. Bauer its representative in Cleveland.

Stephens-Adamson Mfg. Co., Aurora, Ill. has appointed F. E. Dunlap branch manager in charge of conveyor sales and engineering for Michigan with headquarters in the Book Tower, Detroit.

Coal Symposium at Pittsburgh

Fuel experts from the United States, Canada and Europe will participate in a symposium on coal at the National meeting of the American Chemical Society at Pittsburgh, September 7 to 11.

Dr. F. S. Sinnatt, director of fuel research for the Department of Scientific and Industrial Research of Great Britain, will present a paper at a session devoted to the hydrogenation of coal. Dr. M. Pier, research director of the I. G. Farbenindustrie A.-G., Germany, and Dr. T. E. Warren of the Fuel Research Laboratories of the Canadian Department of Mines will be other speakers. Research reports will be read from Pennsylvania State College, the Coal Research Laboratory of Carnegie Institute of Technology, and the U. S. Bureau of Mines. The extent and duration of the nation's oil resources will also be discussed.

A session on the carbonization of coal will be addressed by Dr. R. E. Gilmore of the Canadian Department of Mines, Prof. Frank Douglas of Colorado College, and representatives of the Illinois Geological Survey and the Carnegie Institute. The Carnegie chemists will make public the results of a study on the influence of the rate of heating on the carbonization of several coals. They have published findings on Pittsburgh coal but new data, it is said, make it necessary to modify some of their previous conclusions.

A third session will deal with fundamental knowledge of the mechanism of combustion of solid fuels. Researches by Dr. H. C. Hottel of the Massachusetts Institute of Technology, by Prof. H. F. Johnstone of the University of Illinois and by scientists at the Battelle Memorial Institute, Pennsylvania State College and the Carnegie Institute will be reported.

The constitution of coal will be the theme of a fourth session at which speakers will include Dr. Harold Hibbert of McGill University, Dr. Ernst Berl of Carnegie Institute, Dr. Walter Fuchs of Pennsylvania State College and Dr. Reinhardt Thiessen of the Bureau of Mines.

Briquetting of coal will be discussed at a special session at which E. R. Kaiser of Battelle Memorial Institute and Prof. C. A. Basore of Alabama Polytechnic Institute will speak. Lignites will be the subject of a paper by Dr. Irvin Lavine of the University of North Dakota, and Dr. H. R. Fife of the Carbide and Carbon Chemicals Corporation will describe new research on dustproofing coal.

Dr. William Durand, noted California scientist, who will act as Chairman of the Third World Power Conference meetings, has found an easy solution of the linguistic difficulties presented in addressing representatives of forty-eight nations.

At the opening session of the Conference on September 7, Dr. Durand will make his address of welcome in four languages—English, French, German and Spanish. He speaks all these languages fluently.

Dr. Durand, who is Professor Emeritus of Mechanical Engineering at Stanford University, and a past president of the American Society of Mechanical Engineers, served on the International Commission on Inventions during the War. He is a Fellow of the American Academy of Arts and Sciences, and is the author of numerous books on engineering.

New Ford High-Pressure Unit Placed in Operation

With appropriate ceremonies, the new 110,000-kw G.E. turbine-generator and high-pressure C.E. boiler were recently placed in service in the River Rouge power plant of the Ford Motor Company. The new turbine-generator, similar to another of the same capacity installed in the station in 1930, is the first large unit anywhere in the world to operate at 1200-lb pressure and 900 F steam temperature, the earlier unit having been for operation at 725 F total temperature. With the additional power generating capacity afforded, this station becomes the largest industrial high-pressure steam generating installation in the world, having a total capacity of 325,000 kw.

The new machine, like the first, is a vertical compound unit with the high-pressure turbine and generator mounted directly on top of the low-pressure turbine and generator. Each element has a capacity of 55,000 kw and generates at 1800 rpm. The physical dimensions include a length of 66 ft, a maximum width of 25 ft and a little more than 23 ft overall height from the floor. The approximate weight is 2,000,000 lb. It occupies less than a cubic foot of space per kilowatt of output.

This is the first 1200-lb turbine to go into operation without reheat.

The boiler, designed for 1400-lb pressure 900 F steam temperature, is of the double set bent tube type, fired with pulverized coal and blast furnace gas, and has a rated output of 900,000 lb per hr.

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